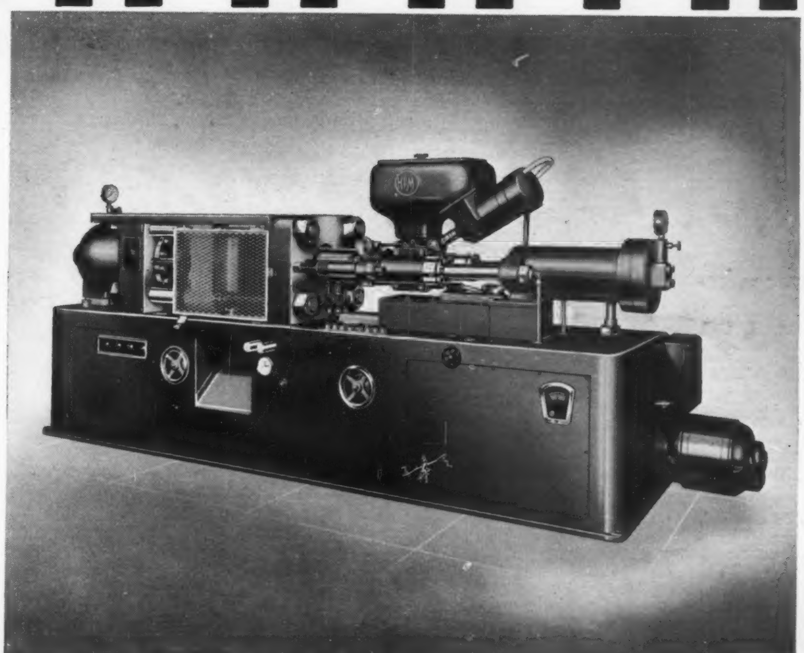
May 1941

MACHINE



DESIGN

In This Issue:

Plastics to Conserve Metals

Designing Snap-Ring Fasteners

Advantages of Torque Motors

Engineer "sticks his neck out!"

YES SIR! An engineer who uses any but the best motors on the machines he designs and builds is "sticking his neck out . . ."

For the wise engineer knows that motors are the *heart* of machine tools. He knows that his machines are *no better* than the motors that power them!

This is the reason why more and more design engineers today are switching to modern Allis-Chalmers Lo-Maintenance Motors!

Low-Maintenance Motors Pay An Extra Dividend!

The story behind these motors is an interesting one—one it will pay you to get acquainted with. For in Lo-Maintenance Motors you get such outstanding features as a stator that's twistless and distortionless . . . a rotor that's indestructible . . . a frame that's made of special high carbon steel for greatest protection.

What's more—you get all-'round performance that's "more than the rating on the nameplate"—an *extra* operation dividend that the design engineer is glad to pass on to the purchasers of his machines.

To get the facts about Allis-Chalmers Lo-Maintenance Motors, call the district office near you...or write Allis-Chalmers, Milwaukee, Wis.

A1388

THE DESIGN ENGINEER OF THIS Stoker-Unit drilling machine has made sure he hasn't "stuck his neck out" on performance. With three Allis-Chalmers Lo-Maintenance Motors powering the machine, he knows he is giving his customers an "extra dividend" in dependable, low-cost motor service.

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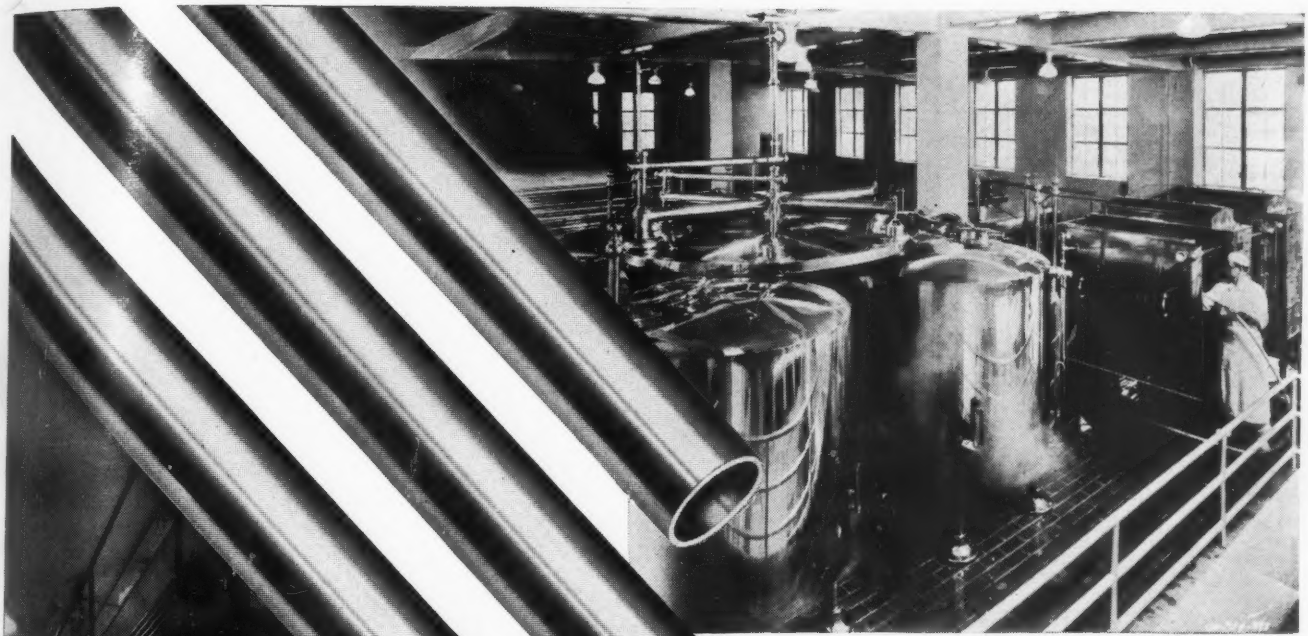

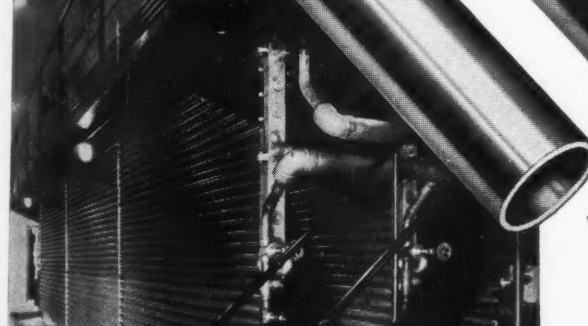

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Topics

PARTICULARLY indicative of what can be accomplished by discussions of mutual problems during times of national emergency is the Machine Tool Electrification Forum recently held at East Pittsburgh. Major design problems were discussed, especially with respect to the defense program, standardization and better performance.

♦

IT IS possible that "punched" data cards for use in business machines may soon become obsolete. At least such is the trend envisioned by a a recent announcement wherein a newly designed machine is capable of sensing information from pencil markings instead of the usual holes or tabs.

♦

OFTEN a design problem could be solved in a better way or a method simplified if water had no surface tension. One solution involves the use of wetting agents in water-lubricated bearings. Also, wetting agents for the reduction of the meniscus to a flat surface for accurate reading of manometers, gages, etc., are often desirable. As well as increasing legibility, they reduce chances of error in readings taken by different observers.

♦

NEW type of intake manifold has been designed for aircraft engines to utilize the heat of the engine. It allows burning of low grade gasoline with the efficiency of high octane fuel. Also, it is claimed that condensation is eliminated at all temperatures.

♦

EXTENDING the use of electrostatic fields for applying abrasives to cloth with sharp points outward for better quality, textile fibers can now be made to stand on end "straight as soldiers." This method is being used in making cloth, to produce patterns without printing.

♦

STANDARDIZATION by the engineering department of Hamilton Standard Propellers for magnaflux tests on highly stressed propeller parts has reduced rejections from as high as 66 per cent to as low as 2 per cent without impairment of quality. For areas subjected to high vibratory stresses, any magnaflux indication parallel to stress lines having a greater length than ¼-inch or any indications located less than 1/16-inch apart in a transverse direction relative to the stress lines is cause for not accepting the part affected. Indications perpendicular to stress lines are causes for rejection in high-vibratory stressed parts. In areas subjected to low vibratory stresses, irregular, heavy magnaflux patterns having a length greater than ¼-inch or straight, heavy, continuous indications following grain lines and having a length greater than 1-inch do not pass.

♦

NOW being completed by the Army Air Corps at Wright Field is one of the most outstanding stratospheric testing laboratories in the country. In it are facilities for providing air testing temperatures down to 67 degrees below zero. This is an equivalent temperature to that at thirty-five thousand feet above sea level. Air is cooled progressively through three stages by being pulled through the first two and pushed through the third by a single fan.

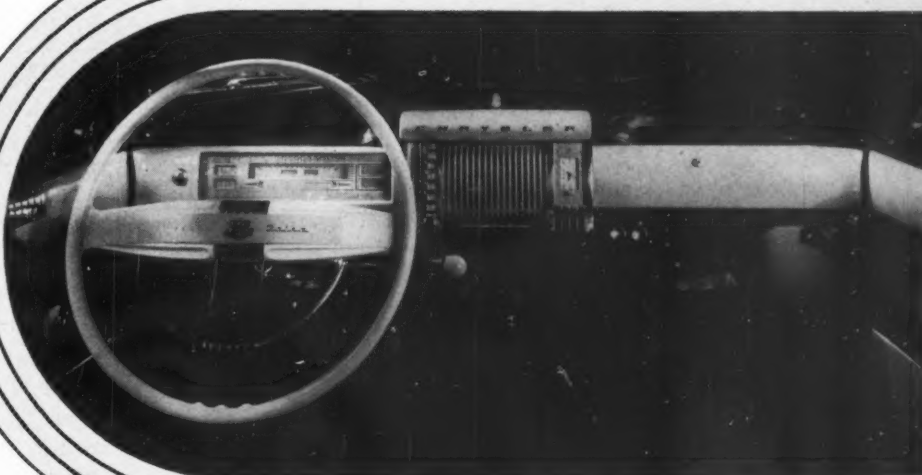
♦

APPROXIMATELY 84½ per cent of the tonage of ferrous foundry products in late years has been cast iron, with 8½ per cent steel castings and 7 per cent malleable castings, it was pointed out recently. Many machine parts which started as cast iron were changed to other materials and have since returned to iron. Automotive camshafts are a good example, and also some crankshafts. Vibration damping was cited as an always-useful property.

♦

BLACK light has been adopted for use in illuminating aircraft instrument panels by the materiel division of the Army Air Corps. Restful on the pilot's eyes, it leaves the cockpit in apparent darkness. Radioactive paint on the instruments and ultraviolet filters on the light sources produce the effect. Maps and charts are also treated for visibilty in black light.

MACHINE DESIGN



By J. Delmonte
Technical Director
Plastics Industries Technical Institute

Plastics Conserve Metals for Defense

RECENT statements by representatives of the O.P.M. urging metal consumers in nondefense industries to use plastics and conserve metals for more vital defense measures will have far-reaching effects. Manufacturers engaging in peaceful as well as warlike activities may be affected. Notwithstanding, such action cannot be said to be unexpected because similar measures have been in effect in England for some time.

Early reactions by metal manufacturers reflect alarm at this turn of affairs. It means that profitable markets may be termed over to other materials, incurring the danger of not recapturing the business when hostilities cease. After the first flurry of excitement, however, a broadminded attitude has

Fig. 1—Above—Automobile dash utilizes plastics. Door trim, steering wheel and knobs also utilize plastics to give a neat attractive appearance

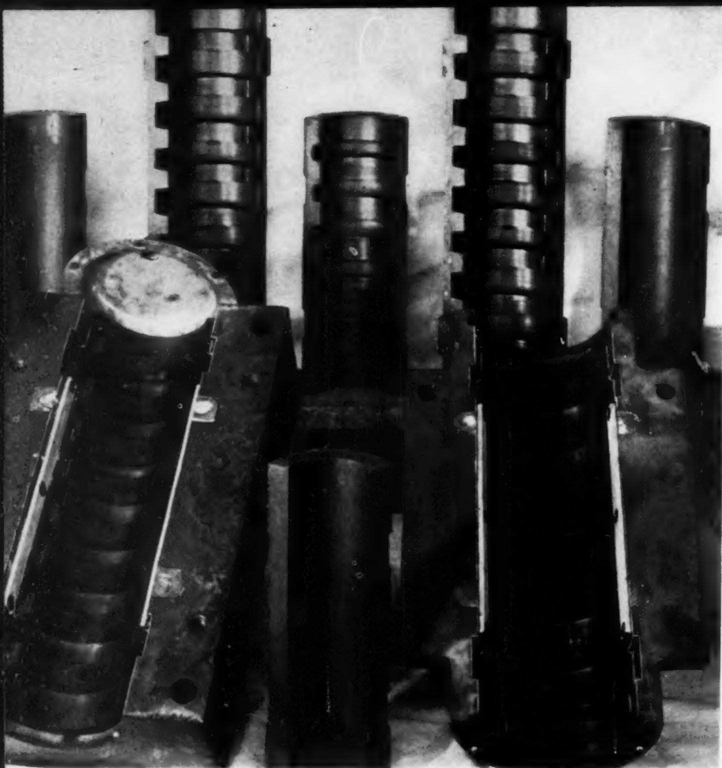


Fig. 2—Plastic bearings have low coefficient of friction and can use water as lubricant. The illustration shows paper mill bearings

been developed. Metal publications are urging co-operation on the part of metals manufacturers in redesigning parts. As far as the plastics industry is concerned, it stands to benefit from the educational aspects of such activity. Manufacturers who refrained from considering "synthetics" before will now turn to plastics and become familiar with their possibilities.

At present there are numerous changes being made to utilize plastics, though in many instances the designs are still on the drafting board. However, increased activity in plastic molding is already evident with the report that activity in this line alone is well over 50 per cent more than that of a corresponding period for last year. In some cases, plastics manufacturers—in anticipation of a shortage of metal—have broken the ice by molding parts of plastics which have in the past been cast aluminum. Aluminum today is taboo in nondefense industries, this metal, as well as magnesium, being urgently needed in aircraft construction.

Where aluminum and magnesium diecastings are to be replaced by plastics, designers familiar with the importance of proper draft, generous radii, adequate wall thicknesses, etc., will have little difficulty in applying similar principles to designs in plastics. They will find it expedient to reduce wall thicknesses somewhat (unless needed for added strength) and to decrease the draft or taper. Because molds for plastics are highly polished, parts may be designed with less draft than equivalent parts in die castings.

While the zinc market is rather stable and manufacturers of zinc alloys claim to have adequate stocks on hand for industrial metal requirements, there is much talk of the growing need for zinc in the manufacture of brass for cartridge and bomb shells. There is a definite possibility that curtailment of zinc consumption by nondefense industries will prevent shortages in the more vital branches of industry. Manufacturers employing zinc alloy die castings are in many cases thinking in terms of plastics in anticipation of the time when they may be required to turn over their production to other materials.

Examination of some of the fields of industrial activity and some of the markets which may be affected by divergence of metals for defense shows how plastics may help to solve the problems.

AUTOMOTIVE: Various decorative appointments, molding and accessories formerly prepared from die castings are being redesigned. For example, one automobile manufacturer reports that five pounds of zinc will be saved on each car in the redesigned models incorporating more plastics. *Fig.* 1 illustrates the use of plastics on a dash.

Experimental efforts in reproducing sections of car bodies in plastics have been successful, though there is no current emergency in sheet steel to expedite this transition. When manufacturers are ready to turn to plastic body parts, it will be because they become convinced that the surface finish is more durable, the material is amply shock resistant, and a considerable saving in weight is possible.

Further automotive parts which may finally be changed over to plastics are: radiator grilles, instrument housings, radiator ornaments, and parts

Fig. 3—Housings for office equipment are dressed up in plastics

in oil and gasoline feed lines.

AIRCRAFT: The aircraft requirements for aluminum and magnesium must be met under any circumstances. It is with this thought in mind that many have been active in developing plastic bonded plywoods for the construction of training aircraft, saving metals for the more important fighter airplanes. Plywoods rendered more durable through the use of synthetic resins show much promise of achieving these ends.

HOUSEHOLD EQUIPMENT: One of the first industries to feel the pinch of metal shortage is household accessories, many of which have long been styled in die-cast zinc and aluminum. For example, vacuum cleaner parts such as nozzles, hose fittings, ends for cylinder type cleaners, etc., are being replaced with plastics. In some of these cases there are previous experiences to govern the design, and there should be not much difficulty in the transition.

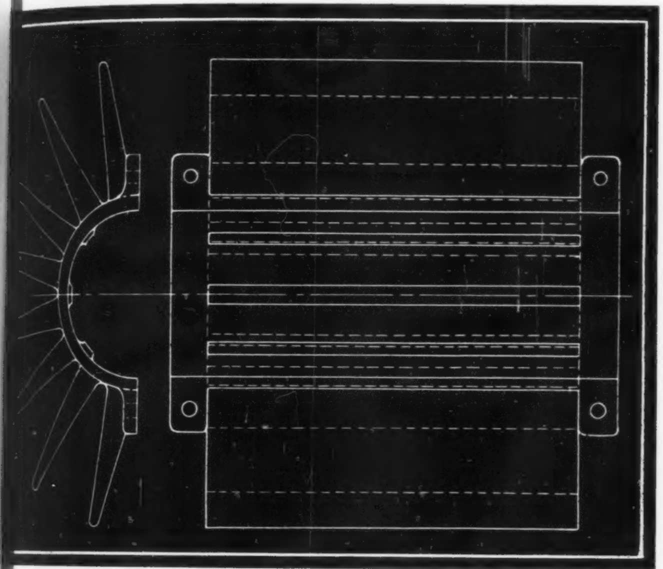


Fig. 4—Fan rotors formerly aluminum are readily converted to plastics

Fig. 5—End bell for a motor frame has steel insert for press-fitting bearing

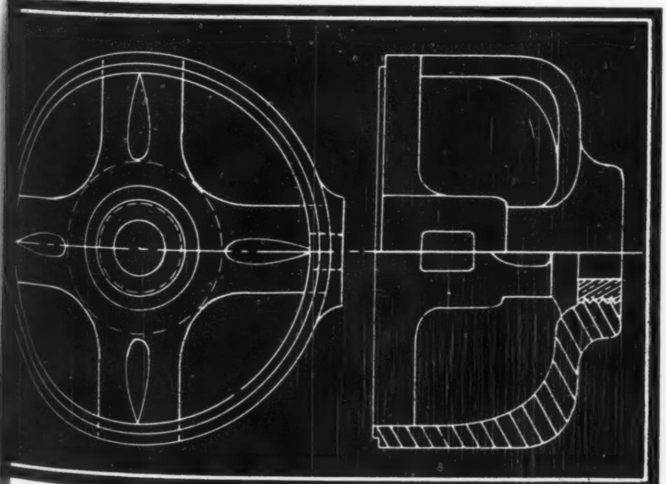


Fig. 6—Gas masks are lightened by utilizing plastics. Outlet valve guards, lenses, and eye pieces are plastics

Fig. 7—Below—Phenolic plastic utilized on viewer-projector is durable, heat resistant and color-fast

Motor housings, frames, handles, stands, gears, and many other parts which are frequently manufactured from die castings of zinc and aluminum are the more obvious parts which will trend toward plastics. The present exigency also will no doubt reawaken interest in nonmetallics for the large quantities of brass bushings and bearings employed every year in household accessories.

BRASS HARDWARES: Faced with an immediate

shortage of brass, hardware manufacturers throughout the country are for the first time seriously turning to plastics to solve their material requirements. We have seen plastic handles for water faucets and laminated plastic seal-off washers but we have not

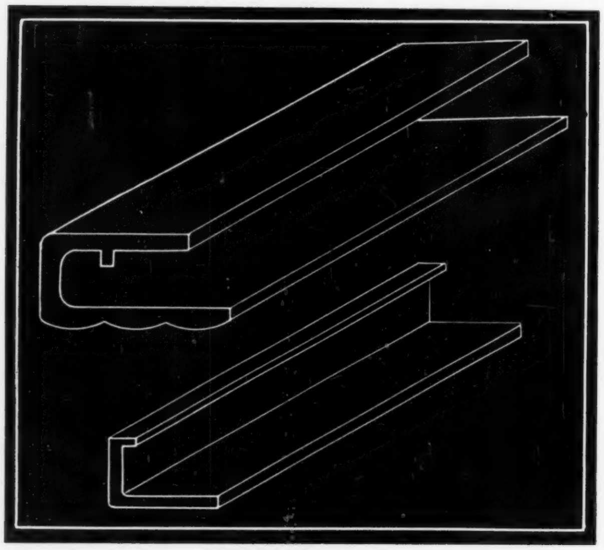


Fig. 8—Typical extruded moldings that are expected to find wide applications for trim

yet seen in production any globe valve bodies, valve stems or pipe fittings in plastics except for special applications such as in the chemical industries.

Business Accessories: Plastics have made notable progress into the field of accessories for business, such as sales register housings as shown by the moldeu ɹrea plastic parts in *Fig.* 3. Interoffice and communication equipment are others.

Refrigerating and Ventilating Machinery: Employing die-cast parts and aluminum parts in many of their assemblies, refrigerating and ventilating machinery manufacturers may find it desirable to change to plastics in many circumstances. For example, the sketch in *Fig.* 4 illustrates a section of an air circulating fan now using aluminum castings which could readily be changed. Refrigerators are now utilizing the decorative as well as utilitarian value in polystyrene injection moldings that are being employed in place of previous metal parts.

Electrical Equipment: There are many small fractional-horsepower motors employing die-cast housings or end bells. Many such parts are being redesigned in plastics, as shown in *Fig.* 5. Plastics, in the case illustrated, equal the physical requirements of aluminum or zinc; also assembly problems are simplified for motor brushes because the plastic is an electrical insulating material. Recognition has been given the limitations of plastics in that instead of attempting to mold a finished opening for the outer race of a ball bearing the designer has employed a steel insert molded into position. The insert has the necessary accuracy for the press fit.

Plastics are contributing to national defense measures in many other ways. For example, *Fig.* 7 illustrates gas mask parts molded of cellulose acetate. Outlet valve guards, lenses, and eye pieces are manufactured by injection molding.

Plastics can be made as strong as metal parts by using thicker sections, stiffening ribs and other expedients. Costs are usually higher but are offset by the colors possible and the beauty of the natural material. Proper choice of material will provide resistance to water, chemicals, etc. Production methods usually involve relatively large quantities because of the costly dies and equipment involved.

When properly applied, plastics may be the answer to many current problems arising from the present divergence of metals.

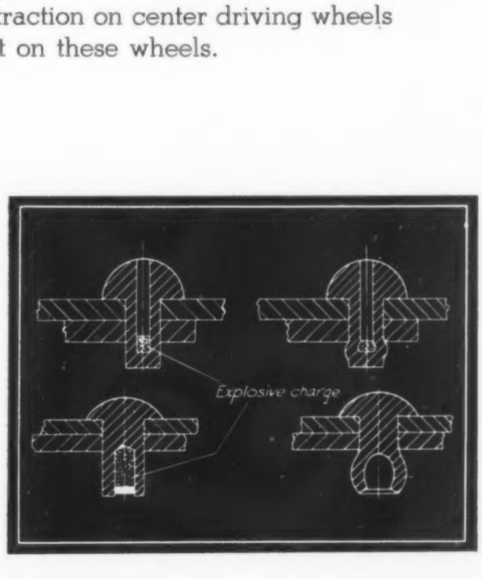
Ford "Blitz Buggy" is designed for reconnaissance duty and is especially capable of travelling over extremely rough ground. Light enough for transporting by a bomber and small enough to conceal easily, it has four-wheel drive, develops 45 horsepower and is equipped with black-out lamps

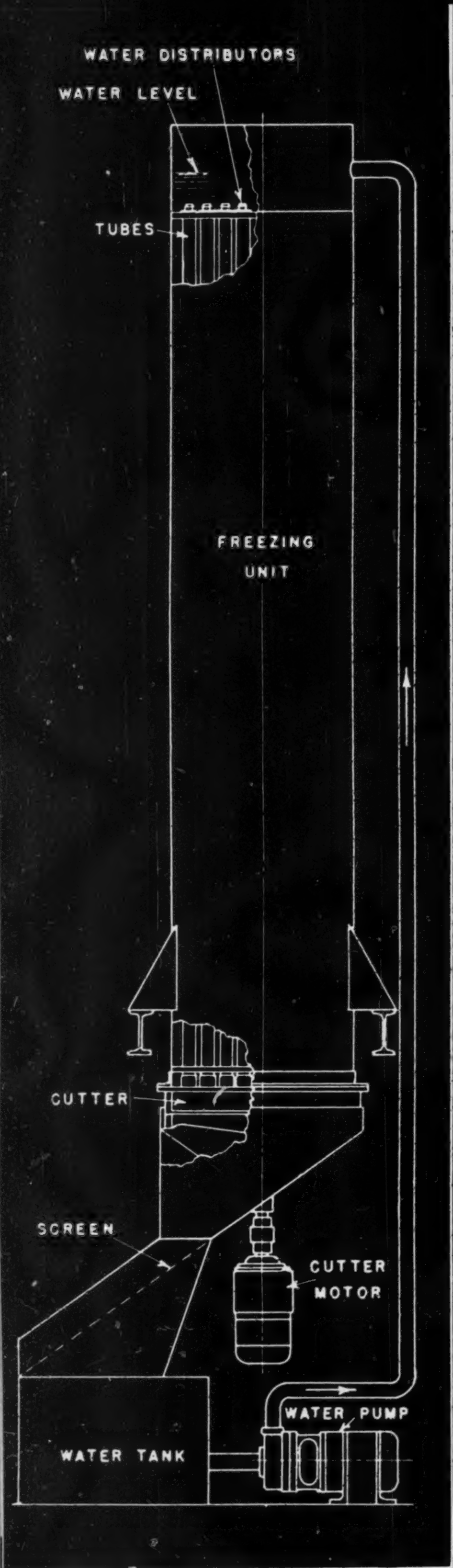
Scanning the Field FOR IDEAS

Pivoting on center wheels, articulating vertically about them and driving through them, this Super-Twin bus involves unique steering and suspension to make the unusual design practical. In this bus steering is conventional to front axle and is reversed to rear by a rod extending along the center of the chassis. Because the body is hinged directly back of the center, a similar hinge joint is required in this rod so the body can articulate without affecting the steering. Booster steering employs two air cylinders.

Body twist is minimized by individual torsion-rod suspension for each wheel. Added stability is gained by wide spring centers and sufficient traction on center driving wheels is taken care of by carrying fifty per cent of the weight on these wheels.

Explosive rivets are again attracting considerable interest because of their promise for fast fabricating and ability for use where backing up is difficult. Two types of rivets are shown, both before and after explosion, in the drawing at right. When the explosive is charged from the rivet head, the closed end of the shaft is apt to be blown off should too large a charge be used. If drilling is from the shaft end this possibility is lessened. The bore is usually one-half the rivet diameter and is three-quarters filled with explosive especially treated against moisture. Charge is exploded through the application of heat to the rivet causing detonation at 150 degrees Cent.





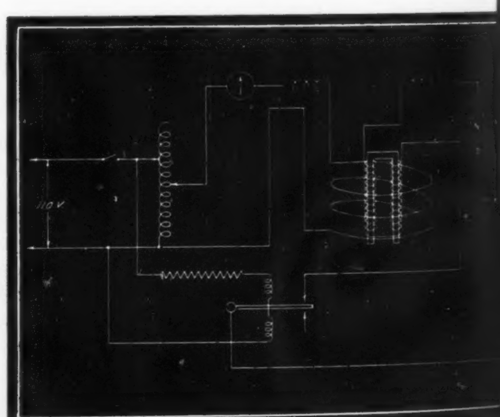
Cutter utilizes gravity feed in the ice machine at left with ice tubes in the vertical position. Water is frozen in the steel tubes by circulating a refrigerant in the shell surrounding them. Either solid or hollow ice is produced, depending on the time of the automatic cycle. Direct application of refrigerant eliminates the use of brine systems and permits high evaporating pressures with resulting economy.

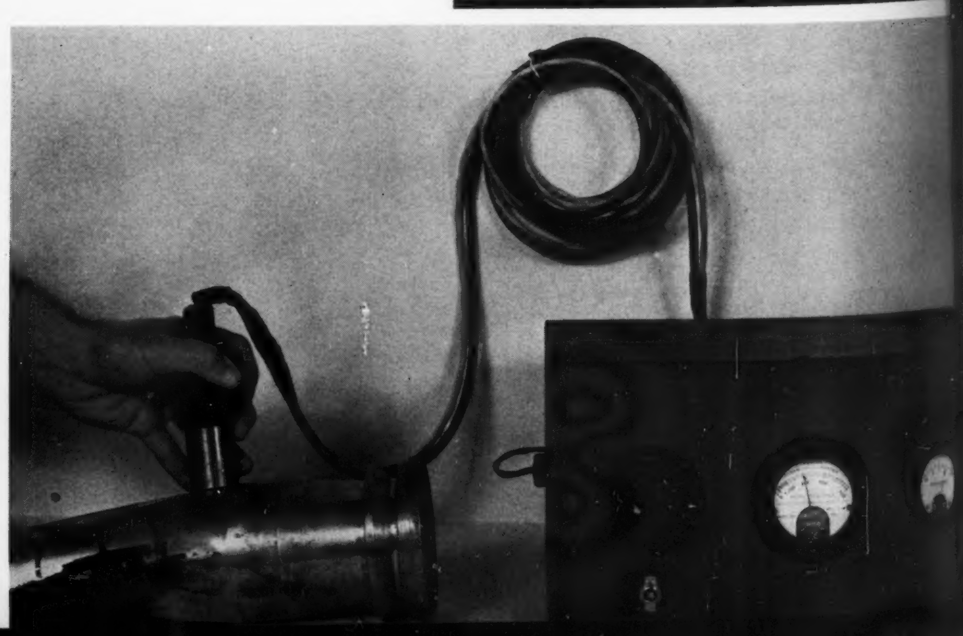
Pumping warm gas through the shell after freezing, frees the tubes of ice which fall by gravity to a rotary cutter and are sized automatically by the position of the cutter head. Water also is admitted beneath the lower tube sheet as an aid to thawing and is frozen in the following cycle.

Flaws in nonferrous and other nonmagnetic materials can be detected with a new eddy-current meter developed by Dr. Gunn at U. S. Naval Research Laboratory. Eddy currents set up in a test piece are compared with a predetermined pattern in a perfect test sample. Apparatus and its circuit diagram are shown below.

In the diagram the power source feeds an autotransformer to provide current for the exciting electromagnet. This current is indicated by the milliammeter in the circuit. In series with the exciting coil, a small auxiliary coil is used with another small coil to neutralize any stray electromotive force which may be introduced into the pickup by misalignment of the parts. The pickup circuit consists of a magnet and coil with an indicating direct-current microammeter. The alternating-current electromotive force induced in the pickup coil by changing currents in the test sample is rectified by a high-speed polarized relay.

Metals with relatively high conductivity such as copper, aluminum, duralumin, etc., can readily be tested. If extreme sensitivity is desired such as detecting flaws in thick sections or if lead, stainless steel or other similar materials are being tested it is necessary to introduce a vacuum tube amplification in the circuit.



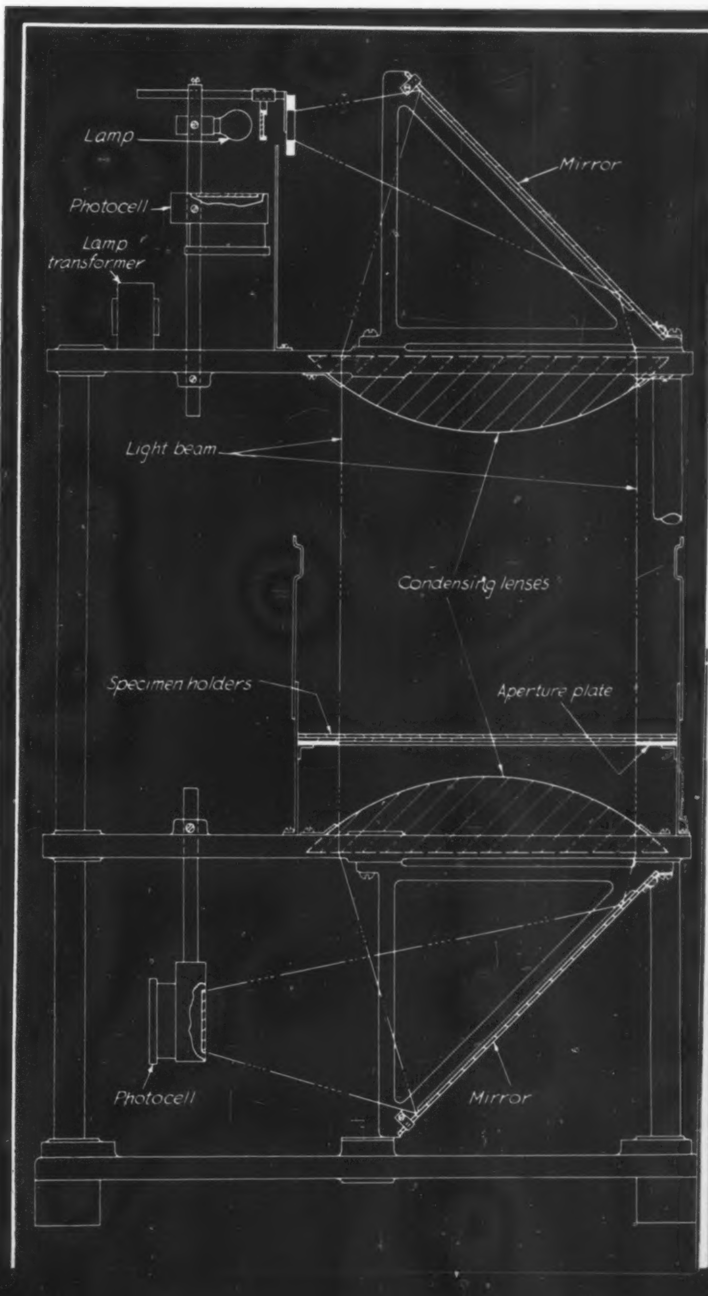


Bimetal mount moves one contact across the face of its mating surface to assure good electrical contact and eliminate the possibilities of glazed contacts. On low-voltage operation especially, glazing might cause insulation of the contacts and necessitate frequent maintenance to prevent operation failures.

Illustration at left is a double exposure showing the extent of movement of the contact due to temperature changes in the bimetal. The right support is a bimetal stamping causing the contact to travel in a diagonal direction. Left support merely reduces vibration. Developed by Westinghouse engineers, this contact has given over fourteen months of service without any maintenance whereas previous types require surface dressing each month.

Photocell comparator measures areas effectively in a unit developed by the American Instrument Co. Utilizing two self-generating photocells in the measuring circuit, balance point between the two is determined by a suspension galvanometer used as a null point indicator.

Optical system employs a light source, diffusing screen, two mirrors and two condensing lenses as shown in the illustration at right. When the system is balanced for 100 per cent light transmission, any opaque object on the specimen holder is measured by a calibrated dial on the galvanometer.



Concentricity is an important requisite for operating parts such as propeller shafts wherein whip or vibration at high rotational speeds is undesirable. Welded steel tubing formed of cold-rolled strip is said to possess this quality to a marked degree. Indicative of this claim is the unusual photograph at left. Produced as a test by loosening the die in a forming machine and allowing the tube to be pushed against the backing die until it wrinkles, each of the concentric rings was produced in this way. Outside diameter of the wrinkles is 3-$\frac{11}{16}$ inches compared to 2$\frac{13}{16}$ inches of the original tube. This amounts to 31 per cent distortion which is 50 per cent more severe than has been obtained in tests carried out on boiler tubes.

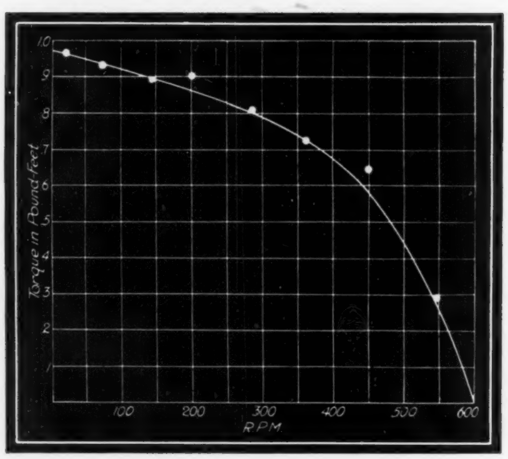
Selectin

By W. R. Hough

Engineer in Charge of A.C. Design
Reliance Electric & Eng. Co.

Fig. 1—Rapid operation of doors on furnace. High-torque motors for short-time rating reduce size needed

Fig. 2—Torque curve of a 3-phase, 60-cycle, 220-volt, 600-revolutions per minute synchronous motor capable of being stalled continuously across full-line voltage



TORQUE motors, for all practical purposes, have characteristics which exactly serve the torque requirement of the application. A distinction, however, should be made: While torque motors may commonly be thought of as always operating at the stalled position, there are in reality many equally important applications where motors designed with special torque characteristics operate around their normal full-load speed. Applications may involve various combinations of torque, rotational duty, stalled duty, and frequent recurrence of these operations. Since the chief demand is for torque, these motors are often rated on a torque and time basis rather than on a horsepower basis.

The general classification into which torque motors fall, insofar as recognized standards of the industry are concerned, is the NEMA Class *D* motor, defined as a high-slip squirrel-cage motor.

Where no appreciable use is made of the synchronous speed of the torque motor, the torque which the motor is capable of developing is alone the important consideration. For example, a 3-phase, 60-cycle, 220-volt, 600 rpm, synchronous-speed open motor built with characteristics to permit it being stalled continuously across full-line voltage has a low-torque capacity with a high-resistance rotor to give an input low enough to be dissipated under continuous stalled operation. In this case, the torque curve is as shown in *Fig.* 2. It will be noted that the input at stalled torque is approximately 140 watts. Since no work is done with the motor stalled, the entire input is dissipated as heat in the motor.

Temperature Rise Limits Stalled Torque

The resulting temperature rise is 50 degrees Cent. on the stator copper. To secure a greater torque would require a proportional increase in watts to be dissipated, with a corresponding increase in temperature rise. If the motor is to be at a standstill most of the time, but with power applied only a part of the time, the resulting temperature rise would, of course, be lower, and it would then be possible to wind the stator to produce a higher torque and still not exceed the rated temperature rise.

If, however, the motor is to operate at some speed rather than to stall continuously, the loss in the motor would be correspondingly less, the heating would be less, and again it would be possible to wind the stator for a higher torque value. Examples of this type of application

pecial Motors

Part V—Torque and High Slip Motors

are shown in *Figs.* 1 and 4. The power input in watts necessary to produce a given value of torque at standstill may be approximated by means of the following formula

$$W = \frac{Torque \times RPM \times 746}{5252 \times K}$$

Using K as .6 (torque efficiency)

$$W = \frac{Torque \times RPM}{4.22}$$

This indicates clearly that where speed as such is not an essential part of an application requirement, the lowest input in watts for a given static torque can be secured by using a low synchronous speed winding. In short, torque requirement, synchronous speed for which the stalled torque motor is wound, motor dimensions and weight, and temperature rise all bear a direct relation one to the other.

Act as Electric Springs

To illustrate a typical application, *Fig.* 3 shows diagrammatically a looper arm, powered by a torque motor, to take up the slack in the steel strip between finishing stands in a steel mill. Making but little actual movement, the motor is stalled continuously, simply keeping the idler roll in contact with the steel strip. A somewhat similar application would be a crane with a torque motor connected to the drum on which is wound the power cable to an electric lifting magnet. Here the torque motor serves simply as an "electric spring" to keep the cable taut.

Where use is made of the synchronous speed of the motor, speed and its relationship to the torque developed are both important considerations. Frequently, such work requires rapid acceleration of a mechanism or a motion up to a certain point, followed by the application of mechanical pressure with the motor stalled. Typical examples of such applications are mechanisms for elevating, clamping, feeding, traversing, counterweighting, and indexing. Operation of valves, chucks, clutches and mechanisms to maintain tension in strip materials, rope or cables are others.

Physically, the torque motor is a high-resistance squirrel-cage motor designed to have the greatest torque-producing ability in the smallest possible overall dimensions. Approximately 40 pounds of motor are required to produce a foot-pound of continuous stalled torque over the range of sizes represented by the integral horse-

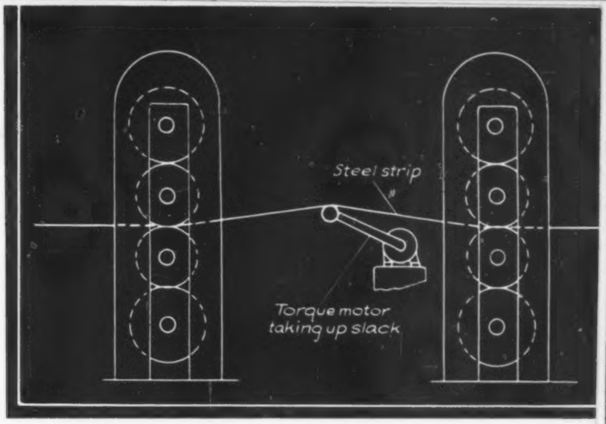
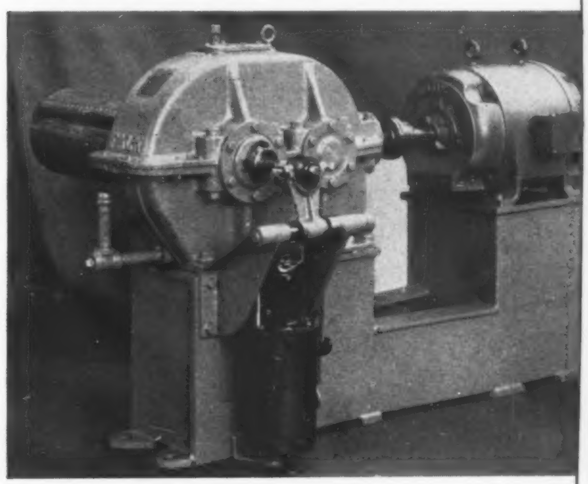


Fig. 3—Looper arm, powered by a torque motor, takes up slack in steel strip between finishing stands in mill

Fig. 4—Speed-torque characteristics of motor on drum coiler are such that a uniform tension is applied to the strip regardless of coil diameter



power motor frames.

From a design point of view, the most important consideration is getting the right relationship between the foot-pounds of torque required for the application and the synchronous speed for which the motor must be wound; in other words, matching the speed-torque curve of the motor to the job to be done. In the design of the normal torque, normal slip motor—NEMA Class *A* or *B*—the rotor resistance is held to a minimum. The NEMA Class *D* motor requires designs employing up to four times the rotor-resistance of a Class *A* or *B* motor for the application involving running at some appreciable speed under a given load, whereas the stalled torque motor may involve designs employing 8 to 12 times the rotor-resistance of a normal slip motor.

There is also a NEMA Class *C* motor which will not be covered in any detail here. Briefly, it may be defined as a high-starting torque, normal-slip motor. Loads with considerable resistance at starting, such

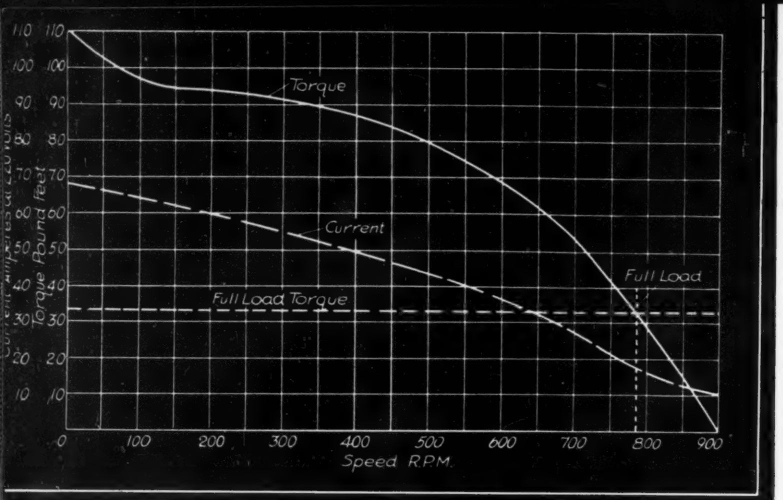


Fig. 5—Typical torque and current curves of a 5-horsepower, 900-revolutions per minute motor suitable for crane or hoist, or auxiliary movements on machine tools. Such motors are for intermittent service only and are rated 15 or 30-minute duty

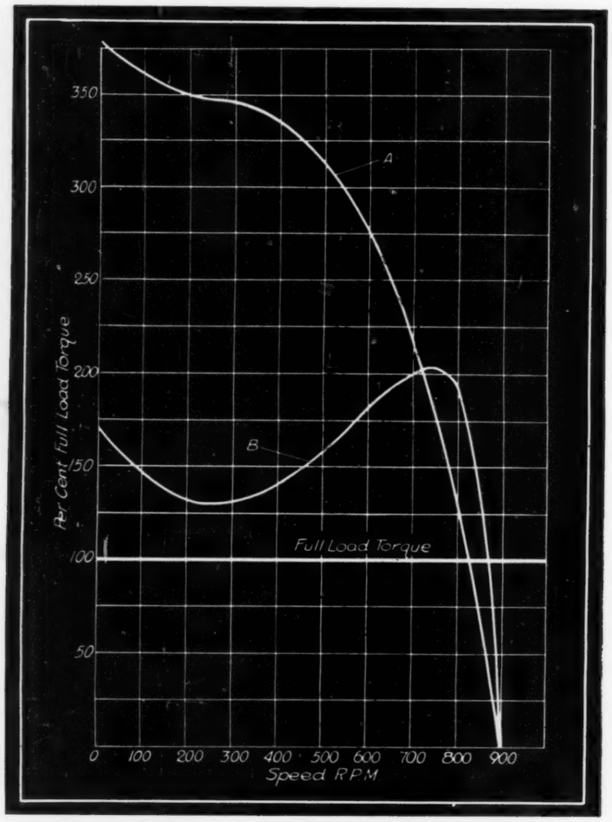


Fig. 6—Speed-torque curves. At *A* is typical curve for a 7 to 11-per cent slip motor, and at *B* the corresponding curve for a general purpose motor

as those of mixers, plunger pumps, large compressors, and conveyors started under load are typical applications for which this motor is suited. To obtain the required starting torque with the normal-torque, normal-slip motor, it would be necessary to over-motor the equipment and thus operate at a lower efficiency and power factor than with a suitable Class *C* motor.

High-starting torque motors (7 to 11 per cent slip) are particularly applicable to flywheel loads, such as punch presses, shears and similar machines. The high slip or speed drop allows the motor to slow down under peak loads which the stored energy of the flywheel then carries. These motors develop approximately 300 per cent starting torque and involve a savings in first cost and improved operation on applications requiring extra torque at starting and having heavy intermittent loads while running. These motors are equally suitable for shears and similar machinery equipped with flywheels.

When the speed of an induction motor is decreased with a load, the energy input increases. Torque motors are designed so that the input to the motor, as its speed decreases, increases *more slowly* than is the case with a general-purpose motor. The peak current loads for a torque motor are therefore reduced over a general-purpose motor for applications of this type.

Provide Economy in Proper Applicaton

First cost of torque motors will be less for given applications than general-purpose motors capable of handling the same work. The torque motor rated for the average running load will have the extra torque needed for starting and for acceleration after impact. When general-purpose motors are applied for such work it is necessary to select higher horsepower ratings than needed for the average running load. In *Fig.* 5 are shown characteristics of a typical torque motor, whereas *Fig.* 6 compares the torque characteristics of a general-purpose motor and a torque motor.

General-purpose motors have advantages, however, when operating conditions require rapidly repeated strokes, such as notching presses, forging and upsetting machines, where the amount of retardation on each stroke must be slight because it is essential that the flywheel be returned to its normal speed in an extremely short time. To permit repeating the stroke without causing the motor to slow down progressively to a stall under the condition of rapidly-repeated strokes requires the use of relatively low slip. The margin of advantage of the 7 to 11 per cent slip motor over a general-purpose motor is lessened on applications which produce over-loads beyond the capacity of the flywheel used.

Allows Use of Smaller Rating

In *Fig.* 7 is shown graphically the current input of the two types of motors while operating a punch press. The solid line shows the current input of a 50-horsepower, 1200 revolutions per minute general-purpose motor. The cycle starts with the engagement of the clutch and the acceleration of the ram downward. Sudden load retards the flywheel until the energy given up by the wheel has accelerated the ram to its full speed. The flywheel, being coupled to the motor, causes the motor to slip to some point below full-load speed and an overload results as shown by the first peak. The second, but lower peak appears as the ram encounters the work and another but lesser retardation occurs. Then the final peak is the result of the work involved in raising

the ram to its starting position.

Indicated by the dotted line is corresponding input of a 25-horsepower, 7 to 11 per cent slip motor performing the same operations. The power input to the general-purpose motor used for this press is almost double that of the torque motor capable of handling the same work.

The curves in *Fig.* 8 show the average losses in each of the motors for which the power input during one press cycle was shown in *Fig.* 7. Load conditions outlined in these charts can be considered typical of punch-press applications, so that it is evident substantial savings can be obtained by applying torque motors designed especially for the work.

Care must be exercised in this connection to distinguish between strictly punch-press applications and others. The punch-press application is one where the time of punching is very slight and there is a relatively longer interval available for reacceleration during which the energy, delivered from the flywheel at a very high rate for a very short time, is restored at a low rate over a longer period of time.

Chart Aids Selection

To arrive at a mathematical derivation of the amount of energy necessary to be returned to a flywheel acting under the influence of a load, it is necessary to know the weight, diameter, speed and amount of retardation or slip of the flywheel. Having these factors and the amount of time between strokes, the horsepower required can be computed by

$$HP = \frac{W \times D^2 \times S^2 \times N}{K \times 5.58 \times 10^{10}}$$

Value of the constant K is determined principally by the reduction in speed of the flywheel and is obtained by observation and check against actual installations. The formula is only applicable where the work is done in less than 1/6 of the total stroke.

When horsepower requirements appear to be small (below 2 horsepower), it is best to check the power required to accelerate the flywheel from rest.

For this condition the following formula should be used

$$HP = \frac{W \times D^2 \times S^2}{5.8 \times 10^{10}}$$

Where the symbols have the same meaning as in the previous equation.

Several applications have been studied where motors appeared to be badly overloaded. In these instances it was found that while each motor was large enough for the running load, overheating resulted from the combination of frequent starting and prolonged periods of acceleration, the result of an extremely large flywheel.

Hoists, small cranes, auxiliary movements on machine tools, and other applications requiring 300 per cent of maximum torque at starting or running under load for short periods of time have designed for them still another class of torque motor. Such motors with 15 to 18 per cent slip and 300 per cent starting torque are for intermittent service only and are rated 15 or 30-minute duty, depending on their enclosure.

A machine tool chuck application also makes use of a motor of the 15 to 18 per cent slip type. Rated

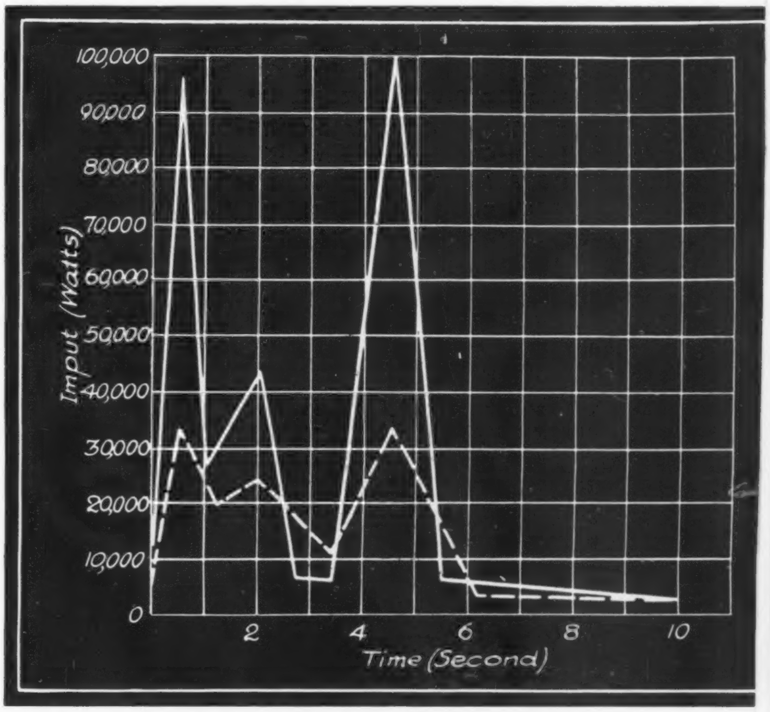
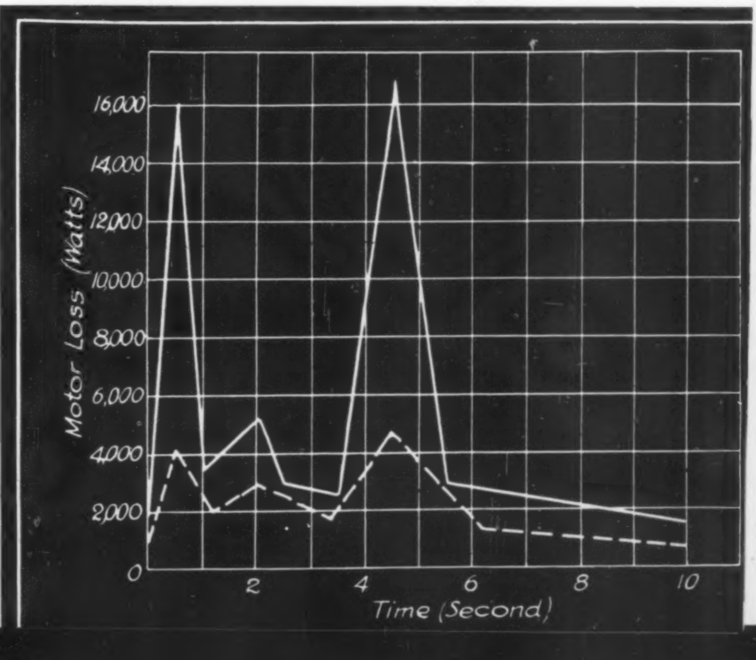


Fig. 7—Comparative current input of general-purpose and torque motors operating a punch press

Fig. 8—Comparative loss curves for general-purpose and torque motors operating a punch press



for 5 minutes duty, the torque motor is excited for the chucking operation, developing 5 foot-pounds torque stalled, after which the power is cut off. To release the chuck it is necessary to develop at least 50 per cent higher torque than that required in the chucking operation itself. This is accomplished by removing the primary resistance in series with the stator, permitting the motor to develop 7½ foot-pounds of torque.

In the field of textiles, an experimental dye jig makes use of stalled torque motors at either end of the jig to pull cloth back and forth through the jig under tension. One torque motor operates to pull the cloth through the jig while the other, with direct current applied to two terminals, acts as a brake and maintains a predetermined tension. The functions of the two motors are simply reversed to reverse the direction of the cloth.

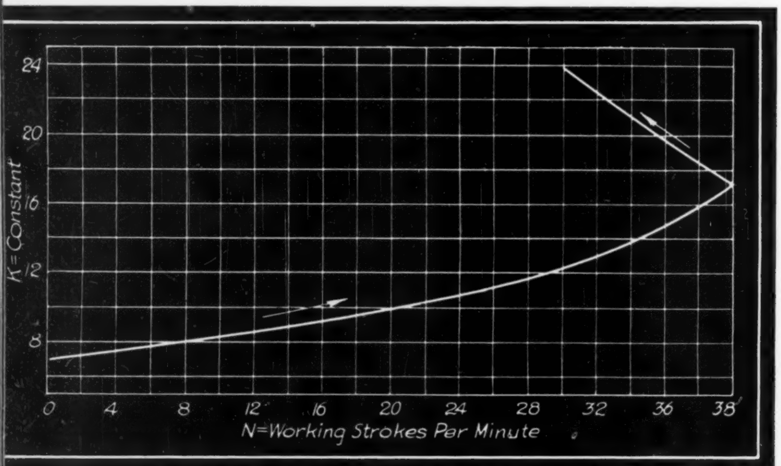


Fig. 9—Curve for constant K used in determining rating of machine with flywheel

Resistance of the connected load of a machine will be zero at starting for machines where the load is not imposed until the machine has come up to speed. However, where compressors, piston pumps, and hoists are required to start under full load, the resistance has to be determined for the value of maximum starting torque in the machine cycle. Then the motor torque delivered in excess of that required to overcome the resistance to rotation, plus starting load on the machine, will be available for bringing the machine up to speed.

Factors Affecting Acceleration

To accelerate the machine, the amount of the torque necessary and the rate at which it must be delivered by the motor will depend upon (1) the moments of inertia of the masses contained in the moving parts and radii of gyration about or with reference to the motor axis, (2) the time allowed for acceleration, (3) the torque necessary to accelerate the load imposed on the machine before full speed is reached, and (4) the torque necessary to overcome resistance to acceleration offered by the friction of the machine parts.

Flexibility and versatile performance possible with torque motors are illustrated in the following interesting applications. To control a slip regulator for a 3000-horsepower wound-rotor motor, a torque motor operates against a solenoid connected to a rocker arm on which is mounted an anode that adds or removes resistance from the driving motor.

In another application a small stalled-torque motor retrieves paper from between steel strips as the steel is rolled up in a coil. The torque motor uses only a few ounces of torque—not enough to risk breaking the paper. However, in this instance, the torque motor must operate well up on the synchronous speed curve.

A particularly interesting application is on an airplane tension hoist on a ship's crane. Here the problem is one of picking up a plane from a heavy sea without yanking out the hook in the plane's fuselage. The torque motor has been applied successfully to solve the problem of maintaining an initial tension on the cable prior to the application of the main hoist motor.

Cable Speed Held Constant

A somewhat similar application of the torque motor has been made on a cable pay-out mechanism. A torque motor with a flat curve and abrupt break-off is used. The speed of the cable over the sheave must be kept at a constant figure. This has been accomplished by designing a torque motor using a skew of several slots of the squirrel-cage winding. Changing this angle of skew so redistributes the flux as to produce the equivalent effect of increasing the rotor resistance and rotor reactance.

To eliminate the maintenance of slipping clutches and the inaccuracy of tension on brass strip as it is wound onto a drum coiler or "blocker," a motor drive with suitable characteristics is utilized as shown in *Fig.* 4. The rolling mill speed is constant. Therefore the blocker has to slow down as the coil builds up on the drum. When the coil gets larger in diameter, the motor's driving torque must increase to maintain a uniform tension on the strip. The speed-torque characteristics of the motor applied are such that as the blocker is forced to run slower with the increase in coil diameter, the torque delivered by the motor increases. A uniform tension is thus maintained on the strip while it is building up from an 8-inch to a full 16-inch coil.

In still another application, a pulp vat consistency regulator utilizes a torque motor to drive agitator paddles the speed of which changes with a change in the viscosity of the mixture. This change in speed, when it drops below a predetermined value, is used to actuate other devices which introduce corrective quantities of ingredients and restore to normal the consistency of the mixture.

Fig. 1—Refinements in design, making for greater simplicity and automaticity, reduce effort in operating calculating machine

Challenges to Design Result in Progress

By John L. Moody

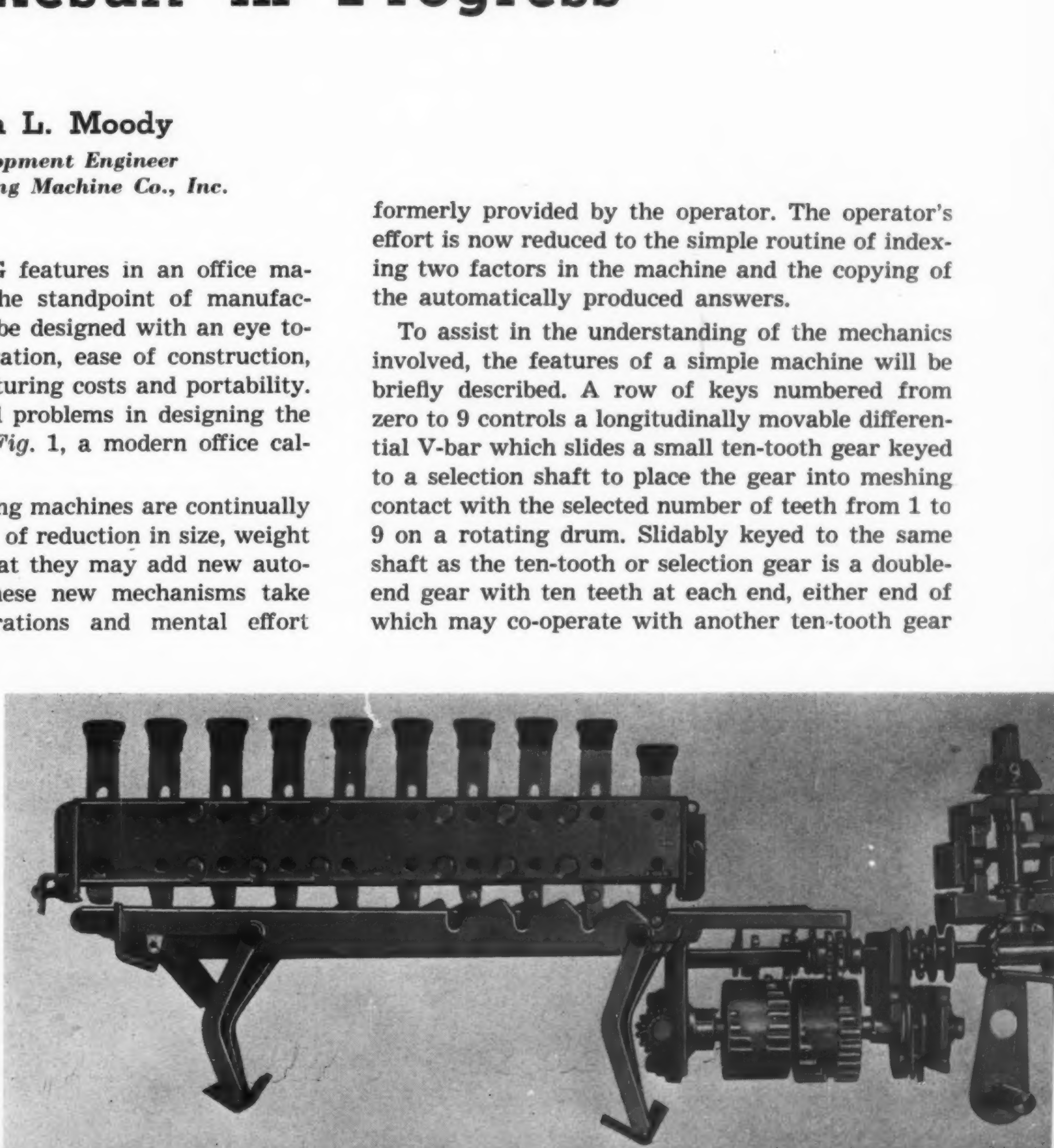
Chief Development Engineer
Friden Calculating Machine Co., Inc.

SPEED-PRODUCING features in an office machine, both from the standpoint of manufacture and use, must be designed with an eye toward simplicity of operation, ease of construction, curtailment of manufacturing costs and portability. Such were the principal problems in designing the machine illustrated in *Fig.* 1, a modern office calculating machine.

Designers of calculating machines are continually working on the problem of reduction in size, weight and number of parts that they may add new automatic mechanisms. These new mechanisms take over the manual operations and mental effort formerly provided by the operator. The operator's effort is now reduced to the simple routine of indexing two factors in the machine and the copying of the automatically produced answers.

To assist in the understanding of the mechanics involved, the features of a simple machine will be briefly described. A row of keys numbered from zero to 9 controls a longitudinally movable differential V-bar which slides a small ten-tooth gear keyed to a selection shaft to place the gear into meshing contact with the selected number of teeth from 1 to 9 on a rotating drum. Slidably keyed to the same shaft as the ten-tooth or selection gear is a double-end gear with ten teeth at each end, either end of which may co-operate with another ten-tooth gear

Fig. 2 — Number of operating parts are reduced by one-half by mounting two drums and two sets of transfer teeth on single shaft



integral with the shaft upon which is fastened a dial also numbered from zero to 9.

When a key is depressed and the drum is given one complete revolution, the same numeral will appear on the dial if the double-end gear is so positioned as to cause a positive rotation of the dial. If the other end of this double-end gear is employed for subtraction a negative rotation of the dial will result.

In the event the numeral is run into the dial twice and adds up to more than 9, a transfer is tripped by a cam fastened to the dial shaft which, when

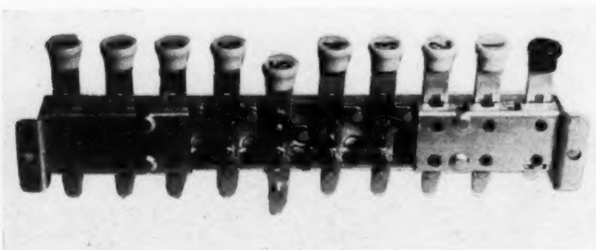


Fig. 3—Above—U-sections are used for positioning in keyboard construction

Fig. 4—Below—Small keyboard obviates duplicate 100-key keyboard

the dial is rotated from zero to 9 or vice versa, moves a ten-tooth transfer gear keyed to the selection and double-end gear shaft immediately to the left of the shaft in which the addition took place. This movement of the transfer gear throws it into the path of a single transfer tooth co-operating with the next order to the left so that one increment is transmitted to the higher orders dial, thereby giving the correct sum or difference of any two numbers which added together are 10 or more.

Simple as this mechanism is, ways and means have been found for improving it to gain even greater simplicity and ease of operation. The greater simplicity, see *Fig.* 2, was effected by mounting two drums and two transfer teeth on a single actuator shaft in such manner as to mesh with the gears on two selection shafts instead of one, thereby cutting the number of these actuator shafts in the entire machine to one-half the number formerly used. A lighter key touch was also effected by mounting two V-bars under each row of keys so that the numeral keys 1 to 5 co-operate with one bar and 6 to 9 co-operate with the other, cutting in half the distance that the keys must throw a single bar when selecting from zero to 9.

After the machine was electrified and the electric carriage shift was added, a means of automatic division was conceived by subtracting the divisor from the dividend in each order one more time than it would go, thereby causing an overdraft or throwing nines into all of the inactive dials to the left of the divisor. This overdraft of the dials is used first to correct the over-subtraction and then to effect a shift of the carriage into the next order where the process is repeated.

Has Only One Clutch

Other interesting facts concerning the electrification of this machine are that there is only one clutch employed in the entire machine, this one clutch being opened by any one of a number of operating bars regardless of whether they are conditioning the machine to effect addition, subtraction, automatic division or automatic multiplication, and that during any automatic operation where carriage shifting is necessary the shifting mechanism is conditioned and a shift takes place in one revolution of the main clutch.

A means of semiautomatic multiplication was also conceived by the adoption of one row of multiplier keys numbering from zero to 9 and designing it in such manner that depression of any one of these keys will cause the machine to revolve the number of revolutions corresponding with the

Fig. 5—Selection levers operated by small keyboard

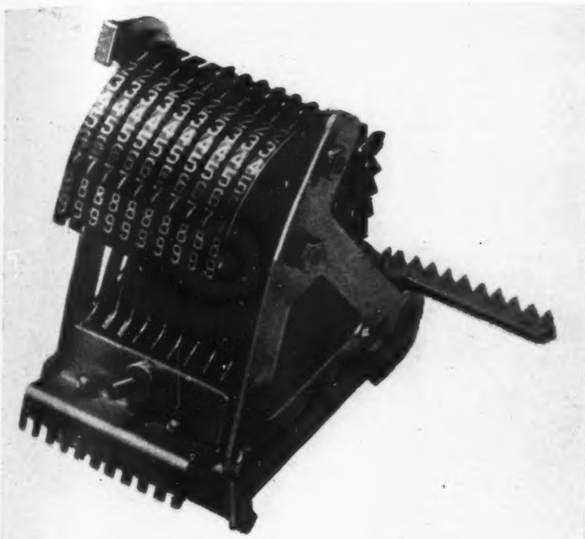


Fig. 6 — Above — Ten-order selection mechanism operates one digit at a time

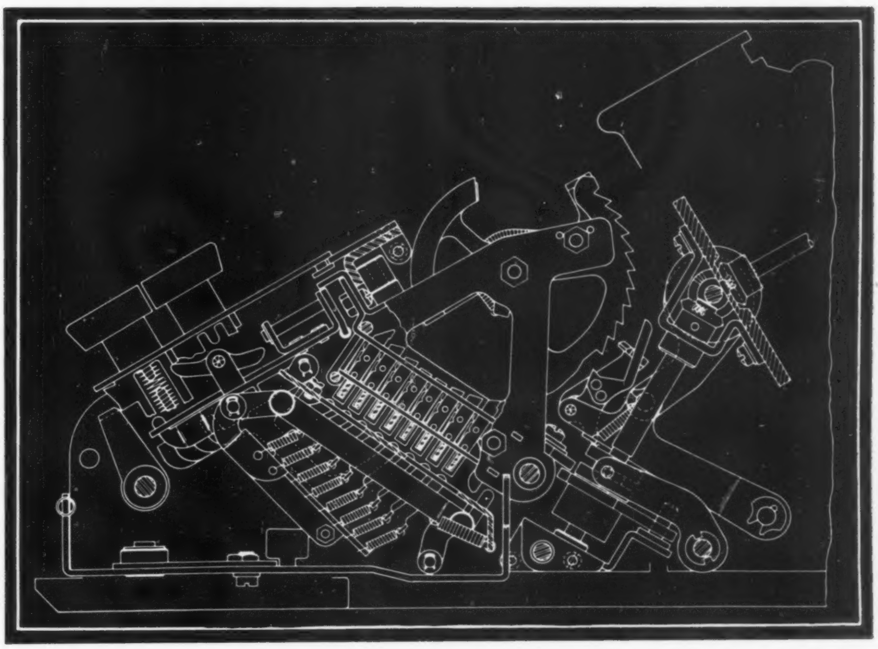


Fig. 7—Right—Close juxtaposition of operating units provides compact machine

numeral on the depressed key and automatically shift over to the next order where it is necessary to depress another multiplication key, if there is more than one digit in the multiplier.

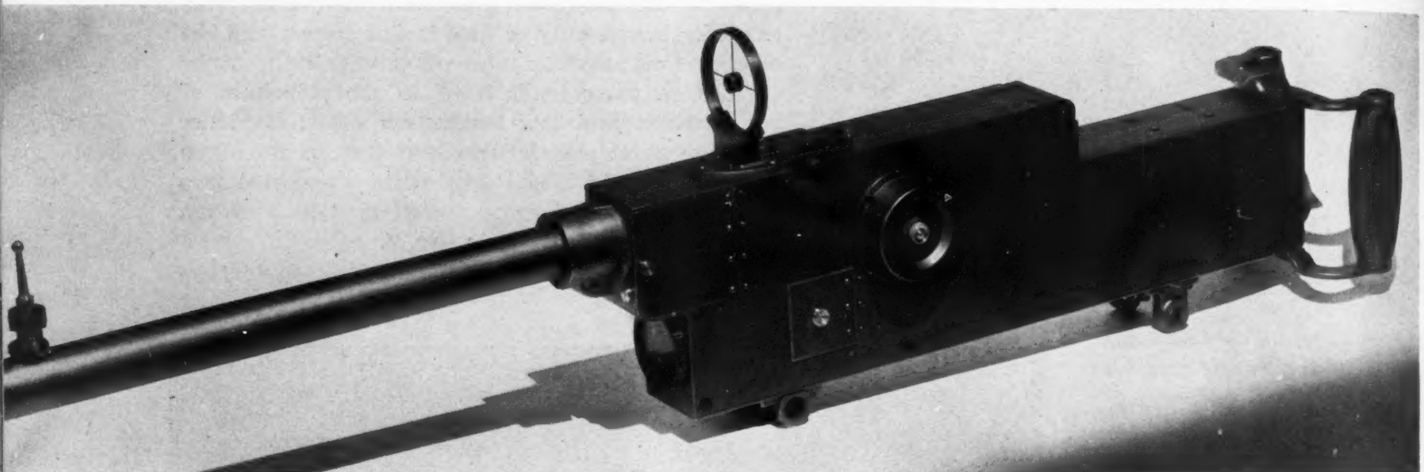
This fully automatic division and semiautomatic multiplication seemed to be the peak in high production mathematical performance, but there proved to be a still greater step in automatic multiplication with the conception of a double keyboard calculator in which both multiplier and multiplicand are set and a multiplication operation (either positive or negative) is effected entirely automatically merely by the depression of a third button (see *Fig.* 1). This could have been accomplished by the adoption of two large 100-key keyboards as shown in *Fig.* 3 which would have necessitated the use of many more parts and made the machine almost twice as large. Since portability and compactness are of primary importance, however, this was accomplished by the conception of a small ten-key keyboard positioned in available space at the immediate left of the large keyboard (see *Fig.* 4) which through a series of selection levers shown in *Fig.* 5, and a suitable escapement mechanism, effects the setting of one digit at a time on a ten-order selection mechanism (see *Fig.* 6) giving it the same capacity as the large 100-key keyboard previously mentioned. This small keyboard, only several inches across, and the co-operating selection mechanism with its multiple production parts lends itself to the principles of compactness and low production costs. Evidence of these principles is also manifest in the close juxtaposition of these units when assembled into the machine to form a complete multiplication operating unit as shown in *Fig.* 7.

Of interest to designers in almost any line where the assembly of multiple parts is concerned is the keyboard construction shown in *Fig.* 3 and the carriage construction in *Fig.* 2. In *Fig.* 3 each row of keys is spaced both transversely and longitudinally by two formed U-sections. These sections are held in place by small projections which fit in slots in the front and rear support bars, and the key stems are supported on the tie bars that fasten this entire unit together. This arrangement is of considerable advantage in economical assembly.

Another novel means of saving assembling costs is the adoption of one long tension spring in each order which serves as the returning force for all keys in that order 1 to 9 inclusive. This eliminates the necessity of assembling 100 springs or one to each key as formerly was done. The carriage is also composed of two U-sections between which the 21

(Concluded on Page 102)

Guns without Bullets



AMERICAN combat planes are reputed to be the best and fastest made. In warfare, however, these planes are merely the vehicle to carry a machine gun into a position to attack the enemy. Effective operation of airplane fire power constitutes the ultimate proof of both plane and personnel. Postponement of fire practice under conditions closely simulating those of actual combat may too easily nullify our superlatively engineered planes and thorough pilot training program.

The machine gun camera illustrated, designed specially by the Fairchild Aviation Corp. for the training of gunners provides an ideal solution to the problem of airplane machine gun practice. Developed to be almost identical in appearance, weight and operation with regulation Browning or Colt guns, these cameras make possible for the first time an analysis of the final combat effectiveness of pilots.

Capable of making from 12 to 16 exposures a second, the camera unit is removable from the "gun" without in any way interfering with its optical adjustments. The lens has a maximum aperture of f3.0 and, with 16 millimeter superspeed panchromatic film, takes usable pictures under practically all daylight conditions.

To check the accuracy of "fire" a glass plate engraved with concentric circles and crosslines is designed into the camera between the focal plane and the rotary disk type shutter. Each frame, therefore, has these circles and lines superimposed.

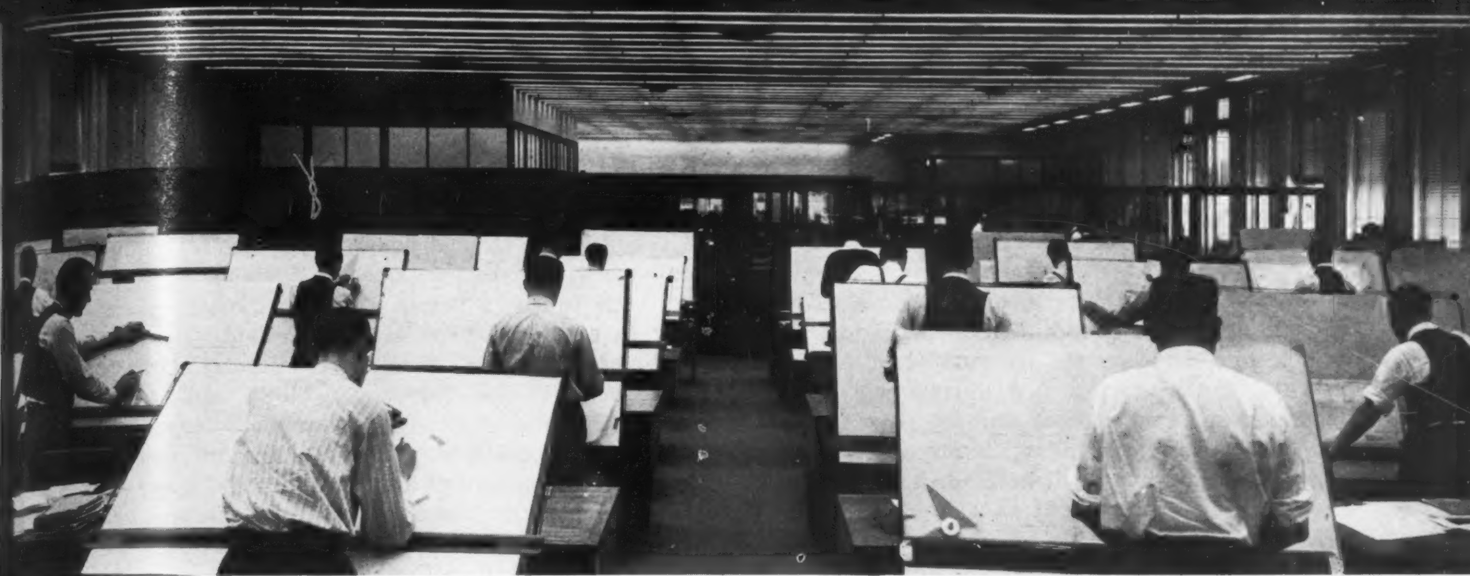
If two planes are in mock combat it is desirable to know which scores the first "hits" on the other. To accomplish this, a system of silvered right angle prisms is used in such a manner that, at the conclusion of each burst, the image of a built-in, split-second watch is projected onto the film. As the prism is moved into the path of the lens by the release of the trigger, four small incandescent lamps are lit by three flashlight-type dry cells. These lamps provide the illumination necessary for photographing the watch.

Further correspondence with actual gun operation is provided by enabling film magazines to be loaded into the camera in the same fashion that cartridge clips are loaded into a gun.

Makes Accurate Analysis Possible

Powered by means of a spring which is wound by a collapsible key on the side of the camera unit case, the camera itself is built up of brass extrusions and stampings to provide proper weight and balance of the unit. The gun case has a durable, corrosion-proof, gun-metal blue finish obtained by a special silver plating process.

With this equipment it is possible for the scoring officer, in addition to ascertaining the time of fire, to determine: (1) Position of the target with respect to the line of fire; (2) correctness of "lead" allowed to compensate for the relative velocities of the plane, target, and bullet; and (3) distance from the plane to the target at the instant of firing, providing the dimensions of the target are known.



Specifying DESIGN DEPARTMENT MATERIEL

By H. T. Pentecost

Part II—Drawing Boards and Tables

EFFICIENT drafting room lighting not only contributes materially to the comfort of the men but also tends to increase speed and output without sacrificing maximum quality and accuracy. The advent and popular acceptance of fluorescent lighting for industrial purposes has come at a time when incandescent lighting was rapidly approaching its maximum strength for year-round comfortable drafting use. This fact is supported by the number of recent fluorescent installations that have been made, a typical illustration of which is shown in the head illustration.

It will be noted that this type of installation may be successfully made regardless of the height or type of ceiling surface formerly considered so important for best reflected light results. This, together with its lower operating costs and its close approximation to daylight, makes it the most effective single step taken to date in the long search for the ideal artificial drafting light.

While better light insures fewer mistakes, especially on the board, it is by no means a cure-all. The human element must still be considered the greatest cause of mishaps in design as in other and more or less mechanized fields. Next to eliminating the obviously careless worker, the reduction of fatigue and its causes is the most di-

Fig. 1—Illustrated arrangement of vertical type drafting boards and consulting tables provides a compact and convenient working unit

rect method of maintaining a minimum of errors. This is an especially timely problem in view of the present demands on designers for more and longer working days. While there is still no substitute for the congenial atmosphere maintained in design departments by intelligent management and careful direction of effort which goes so far in postponing fatigue and its effects, much has been done in controlling its physical causes.

Poor air may well be classed as second in importance to poor light as a major cause of physical discomfort while working. Modern air-conditioning installations are supplying many drawing rooms with fresh clean air in locations where free ventilation was formerly all but impossible.

The type of drafting furniture engineering departments supply their designers is especially important at present when all industry is attempting to utilize to maximum advantage every square foot of existing floor space. For convenience, almost all of the drafting boards now in use may be considered as falling in either one of two types: The older and more common nearly horizontal type, and the newer vertical board.

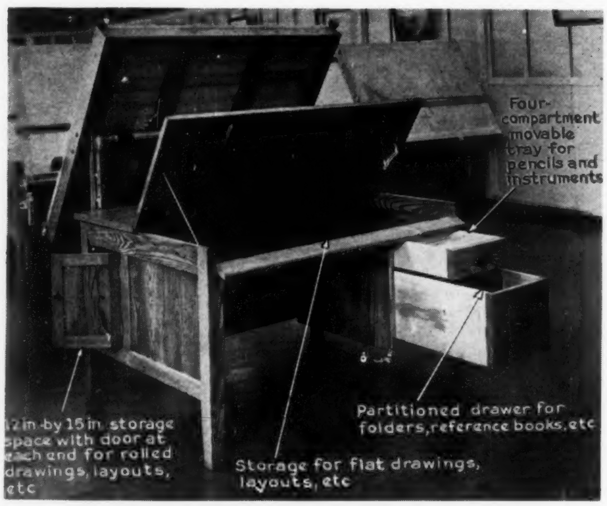


Fig. 2—Consulting table affords desk, filing and storage facilities as well as support for vertical drafting board

By the nearly horizontal board is meant both the flat table top board as well as those tilted at small angles, before which a designer stands or sits upon a comparatively high stool. The dimensions of these units are seldom standard and may range in length from individual tables of four to five feet up to room length, built-in equipment used by several persons at one time. These long boards are especially useful for work on very large drawings. In general, this horizontal style, besides being relatively inexpensive, utilizes the available light to better advantage than the vertical type and is favored by some draftsmen since a number of his tools and references may be kept very close at hand for quick use. For this reason the horizontal table is thought to make possible more rapid work than the vertical board.

By the vertical board is meant that equipment which may be positioned in a nearly vertical manner before which a designer may stand, or sit in an ordinary office swivel chair. Its chief advantage lies in the fact that all parts of the board, and thus the drawings upon it, can be reached with little effort. The maximum length of this type is seldom over five feet so that special provisions for longer drawings must be made at both ends of the board. Because of its position this board, together with the space necessary to seat an individual before it, would naturally require less floor area than the horizontal type.

To match the latter in utility, provision must be made for an additional table area on which the designer may place his various drafting aids and paraphernalia. This is usually done by supplying a "consulting" table containing drawer space which, together with the vertical board, makes the individual drafting unit. Such a unit is shown in *Figs.* 1 and 2. This has been especially designed to meet the majority of requirements of a large company and features a large flat compartment under the consulting table top for storing drawings flat, an instrument and reference book drawer, a standard 10 x 12-inch filing case drawer, a long narrow compartment accessible from the sides or back of the table for storing rolled drawings and paper, and a vertical drawing board mounted on the back of the table in such a manner as to be adjustable in both height and angle. A row of ten of these units, together with special end pieces, will furnish eleven designers with a board and consulting table with a working space between.

Machines Increase Speed

Utilizing the vertical board in this manner requires little more space than the single horizontal board and, of course, much less space than a combination horizontal board and consulting table.

Units of similar design are available with certain additional refinements including ingenious mechanisms which allow the draftsman to bring his work within easy reach by raising or lowering his board at will without leaving either his sitting or standing position. Use of such units, especially if they are also equipped with one of the modern drafting machines, largely offsets the disadvantage of the vertical board's inability to support drawing tools within the immediate vicinity of the working area.

Although there are several different styles of these so-called drafting machines on the market, all have as their purpose a means of positioning calibrated straight edges in such a manner as to make the use of such formerly indispensable draftsmen's tools as T-squares, scales, triangles. and protractors unnecessary. These machines are usually best suited for installation on vertical boards where they perform most efficiently.

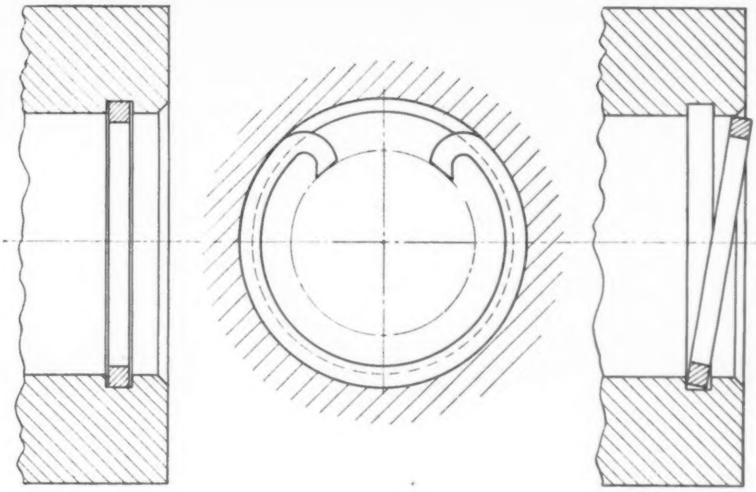


Fig. 1. Snap rings near an external face are readily removable by canting

Designing Snap Ring Fastenings

By Peter F. Rossmann

Development and Research Engineer
Curtiss Aeroplane Division, Curtiss-Wright Corp.

SNAP rings provide an economical and effective means of facilitating machine assemblies in applications where the loading is not excessive. Since snap rings are relatively simple fastening devices, inadequate attention often is paid to their design and specification—resulting in either holding failure or excessive deformation upon installation or removal.

Internal snap rings usually are assembled into grooves near the face of the part in which they are used, as shown in *Fig.* 1. In this case their removal from the groove is accomplished readily with pliers by merely canting the ring after the hooks have been compressed.

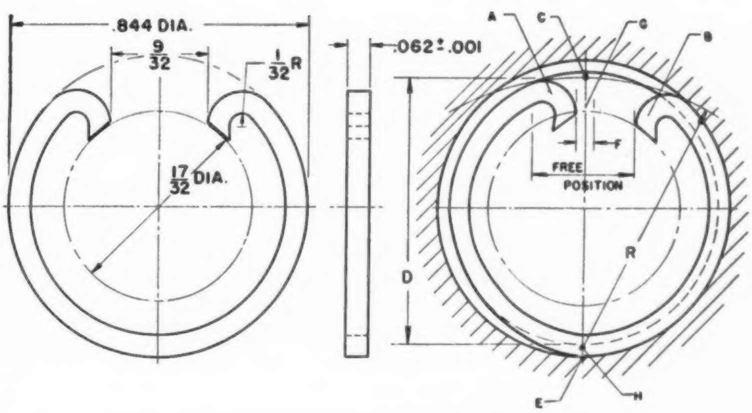


Fig. 2. Because of large deflections necessary for removal, this design resulted in frequent failures

When the ratio of the pitch diameter to the ring cross section is relatively large, the gap between the hooks can be great enough to permit the ring to be compressed sufficiently so that it may be removed without canting. This ideal ratio of pitch diameter to cross section is not always possible, especially if the resulting cross section is too scant for adequate load capacity. That is, if the groove is located deep in a hole, and the groove itself has so much depth in the radial direction that a ring of considerable thickness is

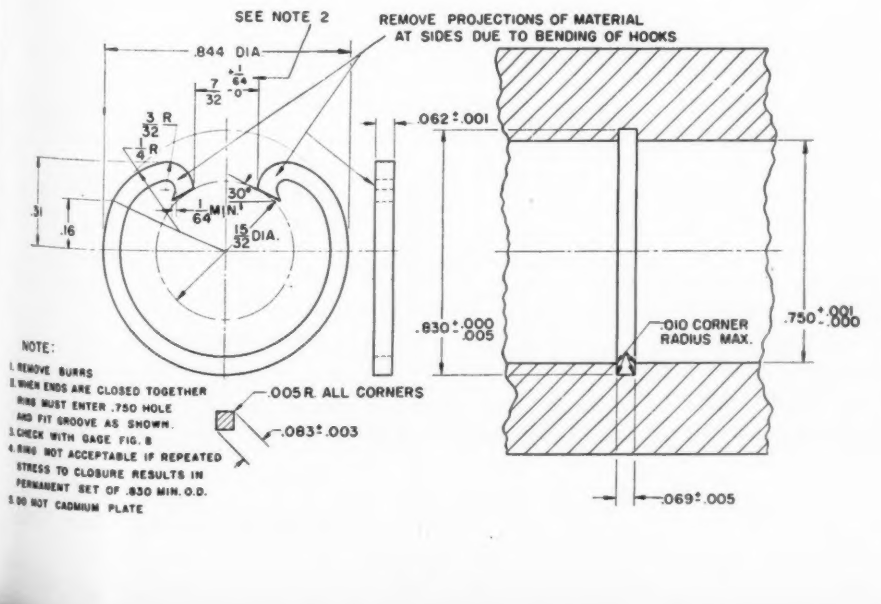


Fig. 3. Inward sweep of hooked ends of ring permits adequate deflection for removal without overstressing

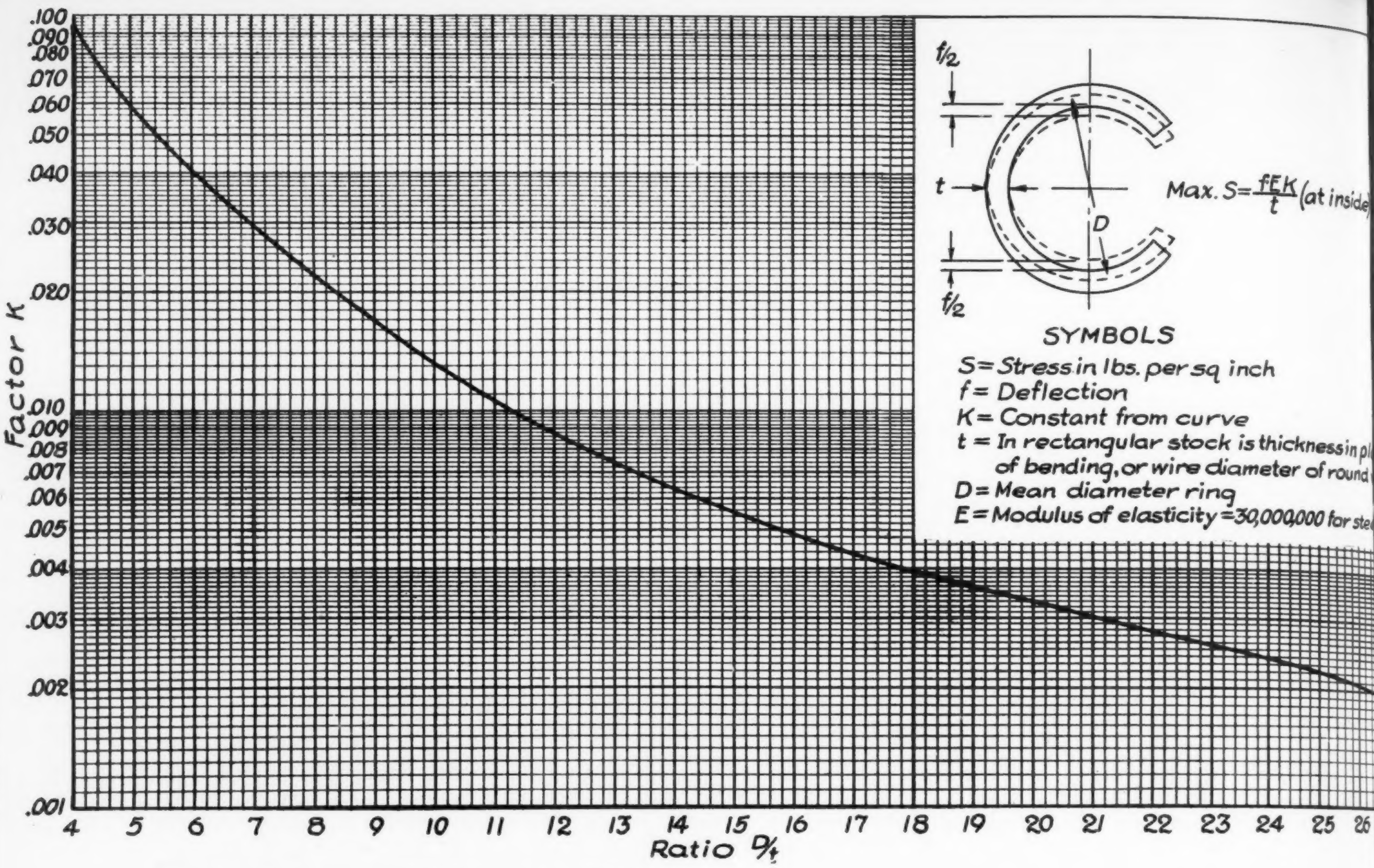


Fig. 4. By determining the stress factor K from the ratio of pitch diameter to radial thickness of the ring, chart facilitates stress calculation in snap rings

required, canting is not feasible and the deflection which must be imposed either to install or remove it may result in a permanent set. In this case, in order to avoid overstressing the ring, careful attention must be given not only to the aforementioned ratio but also to the clearance between the hooks.

Shown in *Fig.* 2 is an original snap ring design that did not meet the safe ratio requirements, thereby necessitating the large gap between the hooks to permit assembly and removal. This design resulted in frequent breakage since, to avoid objectionable set which would result in excessive and undesirable looseness, a critical degree of hardness was attempted to accommodate the high stresses imposed. On the right of *Fig.* 2 is shown the result of a study which was made of the ring profile when compressed. It was determined empirically that the outside surfaces of the hooks A and B revolved through an approximate radius R (actually this is not a true radius). The compressed diameter CE is insufficient to permit removing the ring through diameter D with the hooks compressed to positions shown at F within the safe stress range. Compressing the ring until the hooks contacted at point G, which was necessary in order to remove the ring, resulted in breakage. This breakage was extremely objectionable and expensive, since it became a major operation requiring considerable skill to remove a broken ring located deep in the body. The fact that the hole was only .75-inch diameter added to the difficulties and the exacting service made imperative a ring design consistently uniform in action and free from breakage.

Maximum bending takes place at point H and there is practically no radial deflection near the hooks. Therefore, the ring was redesigned with the hooks A and B having an inward sweep shown in *Fig.* 3. The change in hook design, which did not appreciably influence the contact area of the ring against the face of the groove, shortened the radius R, *Fig.* 2, and permitted compression of the ring until the hooks contacted at G without objectionable set. The causes of breakage were thereby eliminated and the diameter CE sufficiently reduced so that the ring could be easily removed.

Attention is directed to *Fig.* 3 representing the

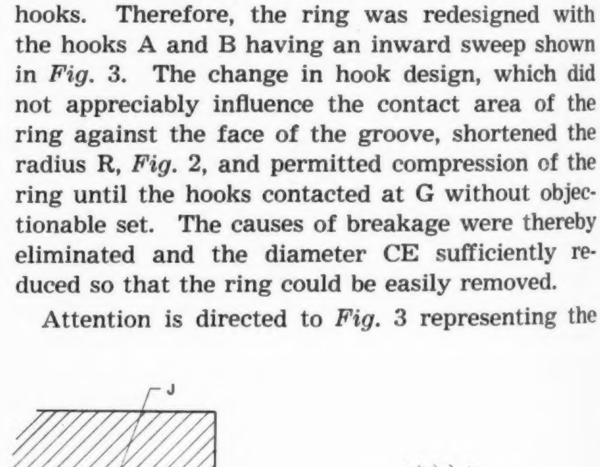


Fig. 5. Use of spring-loaded plug behind snap ring provides safety measure in the event of breakage

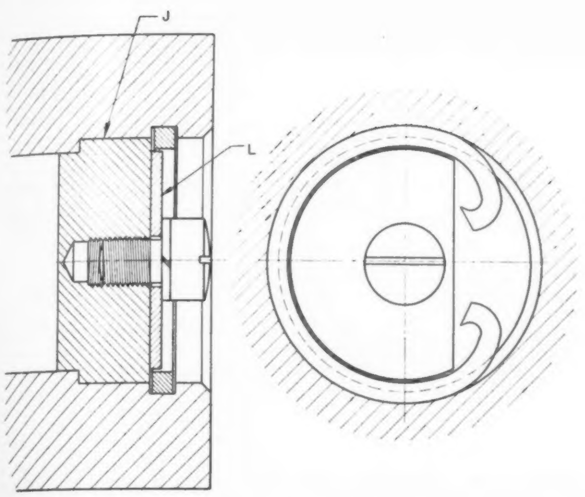


Fig. 6. Special washer fastened to shouldered plug by a cap screw provides an alternative to the spring loaded floating plug used in Fig. 5

final production drawing which indicates manufacturing specifications and wire cross section as well as the groove into which the ring must fit. Specifications in Note 2 of *Fig.* 3 for accurately controlling the ring profile, thickness, and fit in the groove, are most easily checked by use of a gage.

It was also discovered that cadmium plating—even though baking subsequent to plating was prescribed for embrittlement relief—was also conducive to breakage. The plating specification was removed and immersion in oil prior to assembly was substituted. Subsequent tests confirmed that this protection against corrosion was adequate for rings located in end grooves which were not within the valve assembly on which this application was made and therefore were not protected by the valve fluid.

It should be noted that in *Figs.* 2 and 3 the outside diameter of the rings in the free position is slightly more than the inside diameter of the groove in order to provide for some set. The outside diameter tolerance is plus or minus .01 after the inspection compression of the ring. Both rings were made from SAE 1065 steel, the final design being heat treated to 40-45 rockwell C.

Although the foregoing indicates the solution of a specific snap ring design problem, the practical application of the inward sweep of the hooks illustrated in *Fig.* 3 should be useful generally in determining proper and efficient snap ring proportions. Radius R and the amount of inward sweep of the hooks can be determined with reasonable accuracy by large scale layout.

Stresses in snap rings can be readily calculated by using the chart, *Fig.* 4. The stresses so calculated are valid also for rings with hooked ends. Although nonferrous material may be used, in this article only steel is considered.

MATERIAL AND STOCK SIZES: An eminently satisfactory material for snap rings, mentioned in the foregoing, is SAE 1065 steel heat treated to a rockwell hardness of C-40 to C-45.

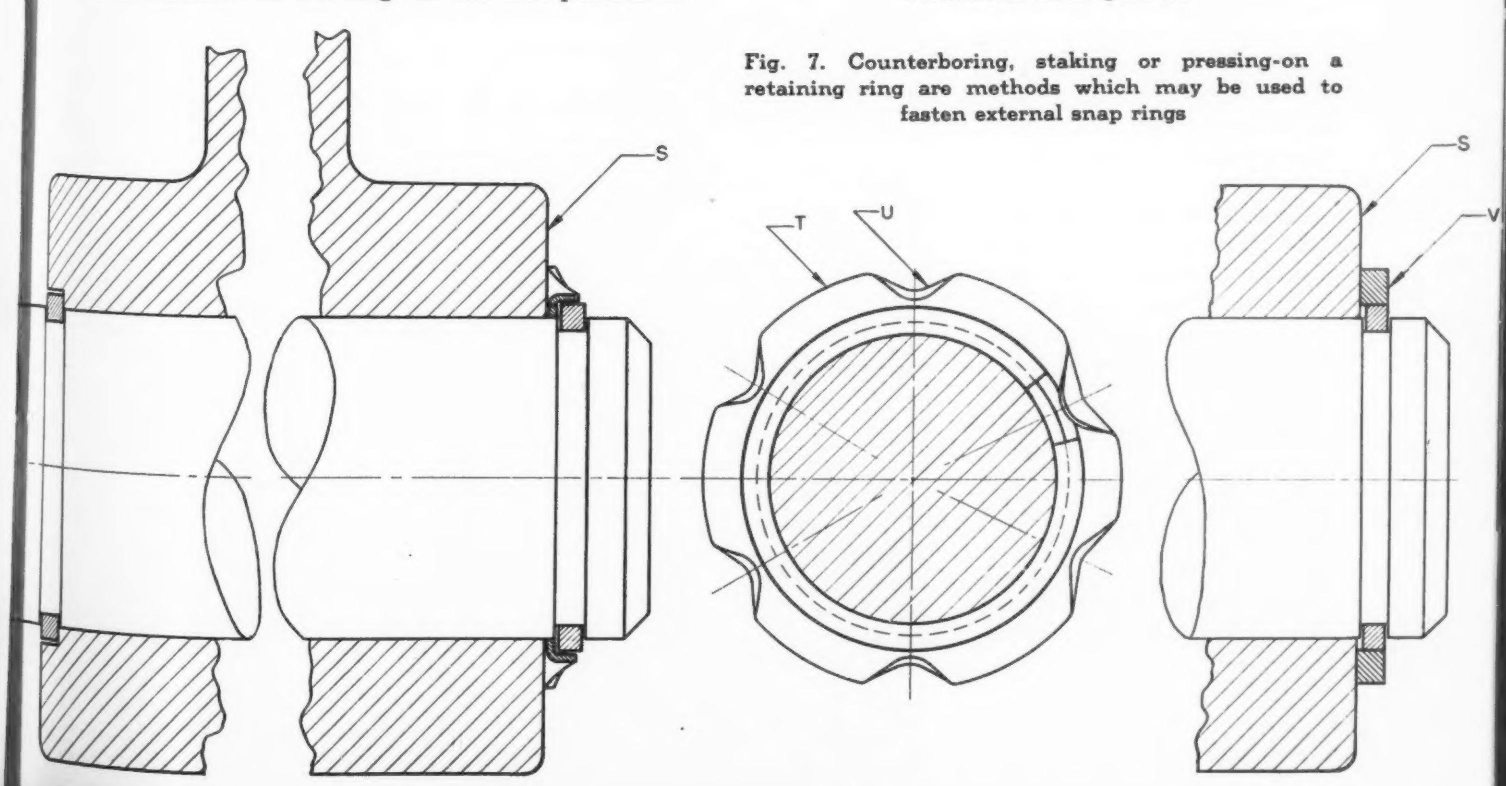
The stock sizes for square wire are .0625, .072, .078, .080, .092, .105, etc. For round wire use music or steel wire gages. Rectangular wire is usually special. Sections having a ratio of thickness in plane of bending to width of about 3 are generally most practical.

DESIGN CONSIDERATIONS: For internal rings, specify an outside diameter that is 1/64-inch larger per inch of diameter than the diameter of the groove. For external rings specify an inside diameter that is 1/64-inch smaller than the diameter of the groove for each 5/16-inch of groove diameter.

The ring should have a constant radius for not less than 220 degrees before giving it an inward sweep as shown in *Fig.* 3. For internal rings the gap between the hooks should be such that, when

(*Continued on Page* 106)

Fig. 7. Counterboring, staking or pressing-on a retaining ring are methods which may be used to fasten external snap rings



Designing

By R. S. Elberty

Electrical Engineer
Landis Tool Co.

IN THE design of complex control circuits, the machine designer will find that optional cycles for the machine may be obtained without much increase in the number of contacts. The change is usually obtained by adding one or more selector switches. A selector switch having two or more contacts is a good example of the type of electric contact that is not independent of all other contacts, since the contacts of the switch are mechanically tied together. A "restrained contact" is a contact that is mechanically, electrically, or magnetically interlocked with at least one additional contact in the circuit. For example, two-pole limit switches, make-and-break pushbuttons, multiple magnetic relays, or electrically interlocked contactors, all have contacts that depend on the position of other contacts used in the circuit.

Safety Features Have Restrained Contacts

Restrained contacts are applied for electrical safety, for electrical sequencing, or for segregating control functions of the machine. Since they are used in complex electrical circuits, a greater amount of skill in their application is required than is ordinarily needed for control circuits using free contacts. Restrained contacts come in many forms, but they can be simply classified into two groups. Contacts that are restrained so that only two positions or combinations are possible may be said to operate under "first order restraint." For example, magnetic relays and snap-action limit switches can have only two positions and operate as first order restraint contacts. "High order restraint" contacts are contacts such as drum switches, sequencing relays and timing relays wherein three or more positions or combinations of contacts are possible.

In *Fig.* 1 is shown an operator's control station for a Landis grinder. Single-pole pushbuttons are used as free contacts. Two of the selector switches are two-position and furnish contacts with first order restraint. The remaining three selector switches have three positions and furnish contacts with high order restraint. Many of these contacts are used for alternate cycles encountered in setting up the machine. The buttons used in the regular cycle are mounted on top of the control station.

Restrained contacts are applied to restrain or limit conditions in otherwise independent circuits. If there is a series of R independent circuits with $(N_1 + 1), (N_2 + 1), \ldots (N_r + 1)$ conditions respectively, the total conditions obtained will be $(N_1 + 1) \times (N_2 + 1) \times \ldots \times (N_r + 1)$. If it is desired to limit or reduce these conditions, it is necessary to add R contacts with first order restraint.

The total conditions will then be $(N_1 + 1) + (N_2 + 1) + \ldots + (N_r + 1)$. Expressed in a

Fig. 1—Operator's control station. Pushbuttons are used as free contacts and selector switches as restrained

rol Circuits

Part III—Restraint of Contacts

simpler way, the conditions are $N_1 + N_2 + \ldots + N_r + R$. However, R contacts have been added to limit the conditions and the total conditions are now equal to the total number of contacts. For first order restraint contacts, the control criterion then becomes

$$F = C = N_f + N_r$$

Where F = Number of electrical functions of a machine, C = Number of control conditions, N_f = Number of free contacts, and N_r = Number of first order restraint contacts.

In any circuit using first order restraint contacts,

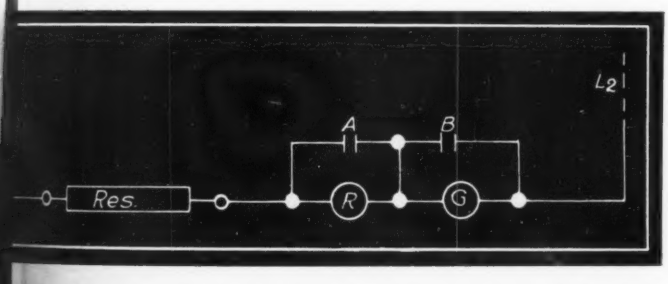


Fig. 2—Because resistance in circuit causes a condition, it is regarded as a contact in applying the criterion

the number of conditions are equal to the number of contacts. The purpose in applying restrained contacts is to limit the conditions obtained from circuits that otherwise would be independent. A properly applied restrained contact will not act as a restraint on any other contact of the same system of restraint. Control circuits that violate this rule will have too many contacts for the conditions that are to be obtained. This condition can be detected by the application of the control criterion which will show more contacts than conditions. The diagram should then be analyzed to find what conditions are duplicated and all contacts that furnish the duplicate conditions can be combined. These will be contacts in the same system of restraint.

Contacts furnishing a high order of restraint will ordinarily add conditions over those obtained from first order restraint contacts. These contacts may take many varied forms and a general development of a control criterion for high order restraint could not anticipate the many possible combinations of this type of contact. However, for general application of the control criterion, high order restraint contacts may be treated as free contacts.

When electrical impedance or other conversion devices are used to modify a circuit to cause conditions, then such devices should be considered as contacts in applying the criterion. For example, if contacts A and B in *Fig.* 2 are free, there are four conditions to be obtained; both lights on, both lights off, red on and green off, and green on and red off. In these conditions the resistance plays the important part of modifying the voltage of the lamp part of the circuit so that there is sufficient voltage for lighting both lamps with both contacts open and so that there is not a dead short with both contacts closed. Since the resistance acts in this way, it actually takes the place of an added contact.

Apparent Exceptions Are Recognized

Engineers applying the control criterion will find many apparent exceptions. For example, the conventional control for motor-operated valves uses two-pole start and stop buttons, two-pole limit switches, and double electrical interlocks. They are wired in a symmetrical circuit on both sides of the contactor coils. While the contacts are at opposite potentials, the added contacts serve the useful purpose of guarding the circuit against faulty operation due to any combination of grounds or shorts in the control wiring that might render a set of contacts inoperative. Except for a dead short circuit, this control will always be in an operating condition. The extra conditions are obtained from the extra contacts, but they do not come into use unless something goes wrong with the wiring.

In *Fig.* 3 is shown a schematic diagram for such a valve. A ground or short in L_1 at points marked $\times$ will make all contacts on the left-hand side of the coils inoperative. Duplicate conditions obtained from contacts on the right of the coil will cause the control to operate correctly. Operation of the valve is of first importance and no fuses or overload devices are furnished. Most motor-operated valves have low-voltage circuits.

Some Single Contacts Act as Multiple

Under certain conditions, single contacts act as two or more contacts. Such contacts can be identified readily since they must operate from more than one source. For example, consider the case of a machine that must be stopped in two or more positions. This might be done by a number of limit switches operated from a single cam. They would then act as contacts with high order restraint. An alternate scheme would be to use one limit switch, and a number of operating cams could be arranged

to stop the machine in the various positions. In such a case the single contact can be counted as a number of restrained contacts, the number of contacts being equal to the number of operating members for the contact.

A common example of a single contact that operates from two sources is a motor-driven timing relay wherein a contact is closed and held by means of a magnet, but opened mechanically at the end of

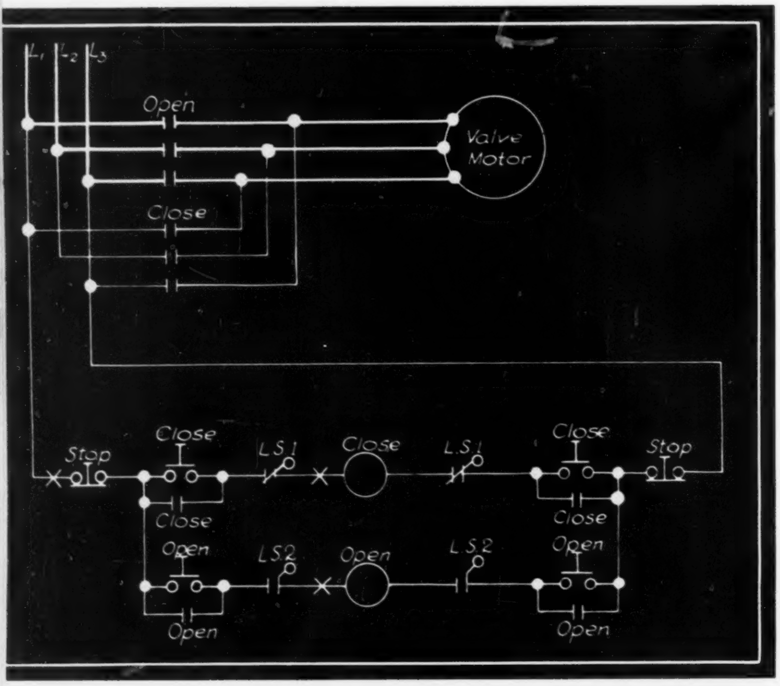


Fig. 3—Schematic diagram for a valve control. Two-pole pushbuttons and limit switches are used, together with double electric interlocks

the timing cycle. This contact is operated from two sources, the magnet and timer cam, and therefore must be considered as two contacts in applying the control criterion.

On series circuits it is possible to arrange the contacts and use a part of them for two purposes. On the machine that is to be stopped in a number of positions, if a number of limit switches are used they can be arranged so that a part of the switches will operate a signal lamp as well as the machine. A number of conditions can thus be duplicated in another circuit through the same contacts. For such a diagram the contacts should be counted for each separate circuit and separate set of conditions. This applies to series circuits only; there is no corresponding exception for parallel circuits.

One Contact Required per Condition

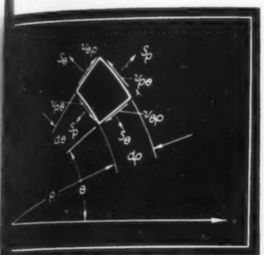
A magnetic relay might be arranged so that one contact will close ahead of the others. The addition of this condition then changes the relay from first order restraint to high order restraint, but in order for the criterion to apply the additional condition must be utilized in the circuit. The control criterion will show if first order restraint contacts can be substituted for high order restraint contacts.

The cases of apparent exceptions to the control criterion always show on analysis that the criterion is met if all conditions are taken into consideration. The greatest difficulty in applying the control criterion is in the complete understanding of the functions that are to be obtained. Here, the machine designer would do well to tabulate the electrical control conditions and associate each condition with the corresponding machine function. This tabulation alone will give a good idea of the complexity of the control, since one control contact is usually required per condition. The types and arrangement of the contacts will depend on the types of control conditions, but starting contacts are usually in parallel and stopping contacts are usually in series. With the control criterion for a guide, the machine designer should follow the rules of good engineering design in working out the elementary wiring diagram. The best control for a machine is the simplest, and good control design starts with the machine designer.

Radiograph Discloses Design

HOW a machine works under actual operating conditions and with the loads encountered in service is demonstrated by this radiograph of a man shaving himself with an electric razor. Taken at 1/1,000,000-second with a Westinghouse X-ray machine, the razor operation can be studied effectively from the standpoint of possible redesign.

Applying Theory of Elasticity in Practical Design



. 28—Plane stress on an ntal prism in polar co-inates. Positive directions of stress are shown

By R. E. Orton
Chief Engineer
Acme Steel Co., Chicago

Part IV—Cylindrical Forces, Shrink Fits, etc.

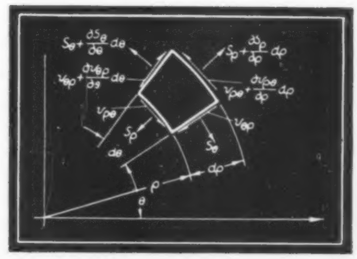


Fig. 29—Stresses acting on the boundaries of an elemental prism. The variation in stress across the prism is shown

PRECEDING sections of this series developed the fundamentals of elastic theory and presented the application of the theory to straight members. The present article will develop the theory for continuous circular members under uniform, normally applied, boundary loads. The stress in these cases is constant over any annular, concentric surface, giving "ring" distribution. This is the condition existing in thick cylinders under uniform external or internal pressures. Its solution permits of the analysis of thick-walled pipes and other circular vessels, cylindrical force and shrink fits, compound guns, etc.

Because equations for the distribution of stress in circular members are simplified by the use of polar co-ordinates, stress and strain components will be presented in this system. In the discussion, ρ will represent the radial distance to any point, θ the angle of the radius from the reference line, S_ρ the normal stress along the radial line and S_θ the normal stress at right angles to the radius, that is, the "tangential" stress. Likewise $v_{\rho\theta}$ and $v_{\theta\rho}$ will represent the respective shear stresses.

Determining Equations of Equilibrium

These stress components are illustrated in *Fig.* 28. As in *Fig.* 1 (Part I of this series) for rectangular co-ordinates, the change in stress across the elemental prism is not shown because the stress at a point is being determined and not the change in stress from point to point. It should be noted that the elemental prism in polar co-ordinates is made up of two radial and two circular faces. Also, the stress directions change as θ is varied.

Stresses on the faces of the elemental prism are presented in *Fig.* 29. The change in values is taken into account as in *Fig.* 3 (Part I) for rectangular co-ordinates. If the forces on the faces are summed in the radial and tangential directions, and quantities involving higher order differentials ignored, the differential equations of equilibrium in polar co-ordinates are obtained

$$\rho \frac{\partial S_\rho}{\partial \rho} + \frac{\partial v_{\rho\theta}}{\partial \theta} + (S_\rho - S_\theta) = 0 \quad \ldots\ldots(61)$$

$$\frac{\partial S_\theta}{\partial \theta} + \rho \frac{\partial v_{\rho\theta}}{\partial \rho} + 2\, v_{\rho\theta} = 0 \quad \ldots\ldots\ldots\ldots(62)$$

Here S_θ has a component in the radial direction and $v_{\theta\rho}$ a component in the tangential. As before, these equations apply only to plane stress or strain, and no body forces.

Any function $\phi(\rho,\theta)$ will satisfy the above equations if related to the stresses by

$$S_\rho = \frac{1}{\rho}\frac{\partial \phi}{\partial \rho} + \frac{1}{\rho^2}\frac{\partial^2 \phi}{\partial \theta^2} \quad \ldots\ldots\ldots\ldots\ldots(63)$$

$$S_\theta = \frac{\partial^2 \phi}{\partial \rho^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(64)$$

$$v_{\rho\theta} = -\frac{\partial}{\partial \rho}\left(\frac{1}{\rho}\frac{\partial \phi}{\partial \theta}\right) \quad \ldots\ldots\ldots\ldots\ldots(65)$$

The condition of compatibility in rectangular co-ordinates, Equation 34, (Part II) may be transformed to polar co-ordinates by

$$\rho^2 = x^2 + y^2$$

$$\theta = \arctan \frac{y}{x}$$

and by the partial derivatives of the function

$$\frac{\partial\phi}{\partial x} = \frac{\partial\phi}{\partial\rho}\cdot\frac{\partial\rho}{\partial x} + \frac{\partial\phi}{\partial\theta}\cdot\frac{\partial\theta}{\partial x}$$

$$\frac{\partial\phi}{\partial y} = \frac{\partial\phi}{\partial\rho}\cdot\frac{\partial\rho}{\partial y} + \frac{\partial\phi}{\partial\theta}\cdot\frac{\partial\theta}{\partial y}$$

to give

$$\left[\frac{\partial^2}{\partial\rho^2} + \frac{1}{\rho}\frac{\partial}{\partial\rho} + \frac{1}{\rho^2}\frac{\partial^2}{\partial\theta^2}\right]\left[\frac{\partial^2\phi}{\partial\rho^2} + \frac{1}{\rho}\frac{\partial\phi}{\partial\rho} + \frac{1}{\rho^2}\frac{\partial^2\phi}{\partial\theta^2}\right] = 0 \quad\ldots\ldots(66)$$

In ring distribution, by hypothesis, the stress is independent of θ, varying only with ρ. Therefore

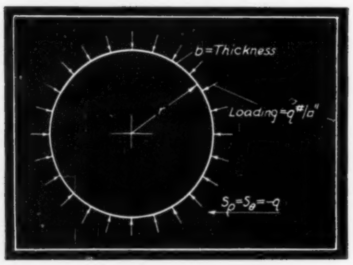


Fig. 30—Disk under uniform pressure

the derivatives with respect to θ are equal to zero. Equation 66 then reduces to

$$\frac{\partial^4\phi}{\partial\rho^4} + \frac{2}{\rho}\frac{\partial^3\phi}{\partial\rho^3} - \frac{1}{\rho^2}\frac{\partial^2\phi}{\partial\rho^2} + \frac{1}{\rho^3}\frac{\partial\phi}{\partial\rho} = 0 \quad (66a)$$

the general solution of which is

$$\phi = A + B\,log_e\rho + C\rho^2 + D\rho^2 log_e\rho \quad\ldots\ldots(67)$$

as may be checked by substitution in Equation 66a. In the equation, A, B, C and D are constants of integration.

Substitution of Equation 67 in 63, 64 and 65 gives the stress equations

$$S_\rho = \frac{B}{\rho^2} + 2C + D(1 + log_e\rho^2) \quad\ldots\ldots(68)$$

$$S_\theta = -\frac{B}{\rho^2} + 2C + D\,(3 + log_e\rho^2) \quad\ldots\ldots(69)$$

$$v_{\rho\theta} = 0 \quad\ldots\ldots(70)$$

This system will embrace all cases of ring distribution of stress. Equation 70 shows the shear $v_{\rho\theta}$ always equals zero. Therefore the principal directions are radial and tangential.

The first shape to which these equations will be applied is that of a solid disk or shaft. Since ρ may be reduced to zero, B and D must equal zero, otherwise the equations would show an infinite stress at the center. If q is the normally applied pressure in pounds per square inch on the periphery $\rho = r$ (see *Fig.* 30), by substitution $C = -q/2$. Substituting in Equations 68 and 69 gives

$$S_\rho = S_\theta = -q \quad\ldots\ldots(71)$$

This is the condition of a hydrostatic pressure. The shear stress is zero and the normal stress is constant in all directions and equal to the external loading.

The general loading for the complete ring shape is shown in *Fig.* 31 where q_1 is the internal pressure and q_2 the external. In this case the constant $D = 0$. Proof of this is beyond the scope of this work. The ring is a case of the "multiply connected body," i.e., a shape that may be cut through from one boundary to the other without dividing it in two. In such case the boundary conditions are insufficient to determine the constants. The condition is similar to a statically indeterminate structure and it would be necessary to investigate the strain relations to establish the other constant. Setting D equal to zero means that the body is not initially stressed.

The constants B and C may be evaluated from the boundary conditions. When

$$\rho = r_1,\ S_\rho = -q_1,$$

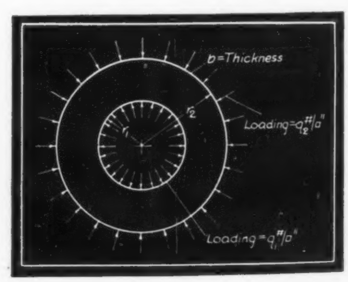
and when

$$\rho = r_2,\ S_\rho = -q_2$$

from which, making the substitution $k = r_1/r_2$

$$B = -\frac{(q_1 - q_2)\,r_1^2}{(1-k^2)},\quad C = \frac{k^2q_1 - q_2}{2\,(1-k^2)}$$

Fig. 31—Ring under uniform internal and external pressures. For boundary stresses see Fig. 32



and substituting these back in the stress Equations 68 and 69

$$S_\rho = -\frac{1}{1-k^2}\left[(q_1 - q_2)\left(\frac{r_1}{\rho}\right)^2 - (k^2q_1 - q_2)\right] \quad\ldots\ldots(72)$$

$$S_\theta = \frac{1}{1-k^2}\left[(q_1 - q_2)\left(\frac{r_1}{\rho}\right)^2 + (k^2q_1 - q_2)\right] \quad .(73)$$

This solution is due to Lame (Paris 1852) and is the fundamental stress formula for the design of thick-walled pipes, simple and compound cylinders at sections away from the disturbance of the ends, etc.

The tangential stress from q_1 acting alone is tension for all values of ρ, and a maximum at r_1. The

stress from q_2 acting alone is always compression and also a maximum at r_1.

From Equation 72, the maximum radial stress is at either boundary and is equal to $-q_1$ or $-q_2$ at r_1 or r_2, respectively. From Equation 73 the maximum tangential stress from both loads acting together is also at either boundary. At $\rho = r_1$, S_θ is given by

$$S_\theta = Kq_1 - (K+1)\, q_2 \quad \text{.................... (74)}$$

where $K = (1 + k^2)/(1 - k^2)$. The value of K may be obtained from the curve of *Fig.* 32. When $\rho = r_2$

$$S_\theta = (K-1)\, q_1 - Kq_2 \quad \text{................ (75)}$$

with K again to be taken from *Fig.* 32.

Conventional analysis of thin-walled cylinders assumes the tangential stress as uniform across the wall thickness. From this assumption the following approximation for the tangential stress is obtained

$$S_\theta = \frac{(q_1 r_1 - q_2 r_2)}{t}, \; \frac{t}{r_2} < .05$$

where $t = r_2 - r_1$ is the wall thickness. In terms of k this is given by

$$S_\theta = \frac{1}{1-k}\,(kq_1 - q_2), \; k > .95$$

The error in the thin-wall equation will depend upon the relative values of q_1 and q_2. For $q_2 = 0$ the actual stress is 2.7 per cent higher at $k = .95$, for $k = .9$ it is 6 per cent higher, at $k = .8$ it is 14 per cent higher, and the error increases rapidly as k decreases. For $q_1 = 0$ the error is k times the above.

"Zero Diameter Hole" Doubles Stress

It is interesting to note from the foregoing that, for internal pressure only, the stress at r_2 is exactly the value of the pressure less than that at r_1. As r_2 increases without limit the stress at r_1 approaches the numerical value of the pressure rather than zero as given by the thin-wall formula. For external pressure only the stress at r_1 is exactly the value of the pressure numerically greater than that at r_2. As r_1 is reduced to zero the stress approaches twice the pressure (rather than one times as given by the "thin wall"). From the previous analysis the stress in a solid disk was found to be equal to the external pressure. Therefore the addition of a "hole of zero diameter" at the center will double the stress.

As previously developed, in ring distribution the principal directions are radial and tangential. Therefore the principal stress difference, Q, is given by $S_\theta - S_\rho$

$$Q = (K+1)\,(q_1 - q_2)\left(\frac{r_1}{\rho}\right)^2 \quad \text{........... (76)}$$

It should be pointed out here that the theory of failure adopted will have an appreciable effect on the design of this cylinder. If the maximum shear theory is used, Q will be the "equivalent" failure stress, that is, the stress equivalent to that prevailing in a simple pull test. For $q_2 = 0$ the ratio of Q to S_θ at the inner boundary is $2/(1 + k^2)$, the limit for which is at $k = 0$ and equals twice S_θ. A

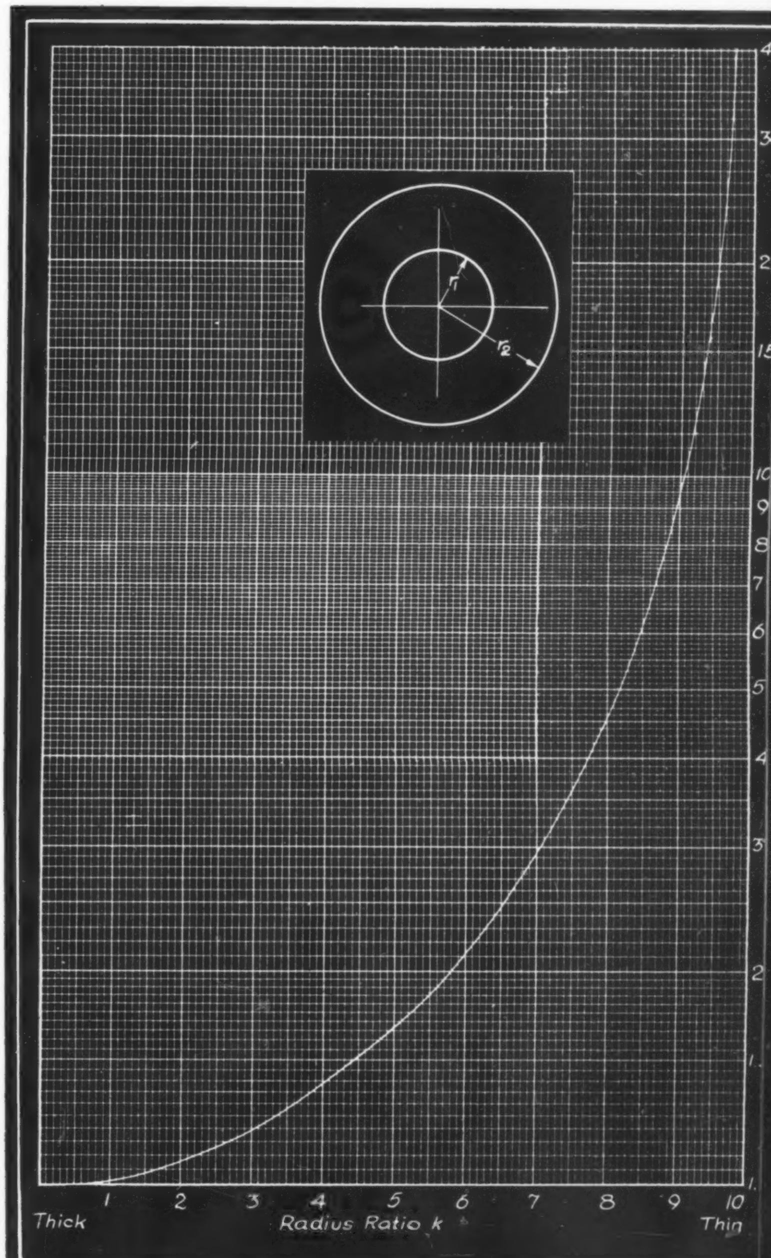


Fig. 32—Boundary stresses in thick cylinders. Curve gives cylinder stress constant K in terms of radius ratio k. Conventional thin-cylinder equations should be used for k greater than .95

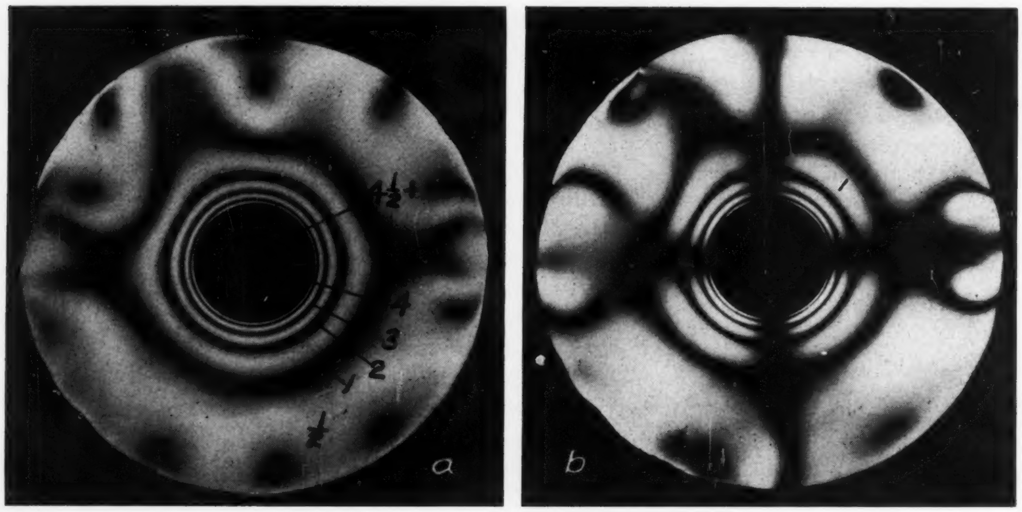


Fig. 33—Photoelastic pictures of a ring under external pressure. At *a* is shown ring under circularly polarized light and at *b* under plane polarized light. Distortion of first order fringe is due to a small break in the continuity of the loading

more elaborate discussion of this subject is reserved for a later article on failure theories.

The principal stress sum is given by

$$S \;\; + S_\theta = (K-1)\, q_1 - (K+1)\, q_2$$

and is constant throughout the ring. By Equation 26 then, the unit strain in the Z direction is constant throughout the ring and plane sections remain plane. Therefore the condition is one of plane stress regardless of the length of cylinder, provided only that there is no endwise restraint. The change in the Z direction is obtained by multiplying ϵ_z from Equation 26 by b, giving

$$u_z = -\,[\,(K-1)\, q_1 - (K+1)\, q_2]\,\lambda\, \frac{b}{E}$$

Timoshenko reports the investigation of G. B. Jeffrey (London, 1921) on the effect of eccentricity of the bore on cylinders under internal pressure only. S_θ is a maximum at the inside boundary at the thinnest point and is given by

$$S_\theta = \left[\frac{1 + (3 - \beta^2) k^2 + \beta^2 k^4}{(1 + k^2)\ (1 - \beta^2 k^2)} \right] q_1 \ldots\ldots\ldots (77)$$

where $\beta = 1 + (e/r_1)$, e is the eccentricity and e/r_1 is limited to ½. The effect is not particularly large on thick-walled vessels. For example, if $e/r_1 = 10$ per cent and r_2 twice r_1 ($k = ½$) the increase in stress is only 5 per cent. On thin-wall vessels the ratio of the increase in the stress to the stress without eccentricity is given approximately by

$$\text{Stress increase} = \frac{k^2\,(\beta^2 - 1)}{1 - \beta^2 k^2}$$

In *Fig.* 33 is shown a photoelastic picture of a ring under an external load, *a* being in circularly polarized light and *b* in plane, axis vertical. The distortion of the first order fringe is due to slight imperfections in the loading. This error has little effect on the fringes near the center. The small spots near the outer boundary are initial pattern, made apparent by the low fringe order. Appearance of a departure from a perfect circle of the outer diameter is due to a rubber strip through which the load was applied.

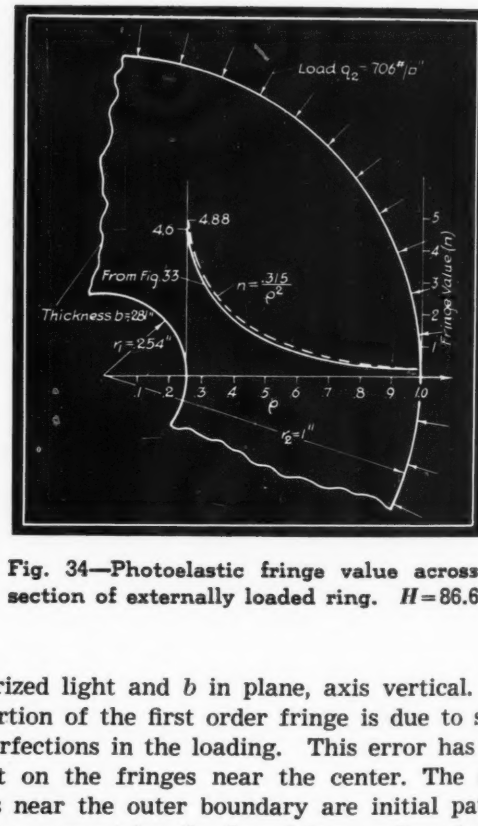


Fig. 34—Photoelastic fringe value across section of externally loaded ring. $H = 86.6$

From Equation 76, the photoelastic fringe value under external load only is given by

$$n = \frac{(K+1)\, b q_2}{H} \left(\frac{r_1}{\rho} \right)^2 \ldots\ldots\ldots\ldots\ldots (78)$$

Dimensions of this model are 2-inch diameter by .508-inch hole by .281-inch thick. $H = 86.6$ and $q_2 = 706$ pounds per square inch. Substituting in Equation 78 gives $n = .315/\rho^2$. This gives $n = 4.88$ at inside, picture shows a little more than 4½,

or about 6 to 7 per cent low. In *Fig.* 34 the value of n from Equation 78 is plotted in the dotted line against the fringe values from *Fig.* 33, in the solid line.

In general it has not been the intention to include analysis for deformation in this series. An exception must be made, however, in the case of the ring to analyze the stresses in compound cylinders. In force and shrink fits the interference of the mating parts compels a distortion of each member. By strain analysis the loading corresponding to this distortion is determined and from this the stresses may be computed by the equations previously derived.

If u_ρ represents the radial movement of a point and u_θ the tangential, the unit radial strain is given by

$$\epsilon_\rho = \frac{\partial u_\rho}{\partial \rho}$$

Since the radial strain elongates the arc at the point, the tangential strain depends partially upon the radial, the unit strain being given by

$$\epsilon_\theta = \frac{u_\rho}{\rho} + \frac{1}{\rho}\frac{\partial u_\theta}{\partial \theta}$$

Hooke's law for plane stress is given in polar coordinates by

$$\epsilon_\rho = \frac{(S_\rho - \lambda S_\theta)}{E}$$

$$\epsilon_\theta = \frac{(S_\theta - \lambda S_\rho)}{E}$$

In ring stress distribution there is no tangential displacement at any point and therefore $u_\theta = 0$. Substituting this in the values of ϵ_θ and equating gives for the radial strain

$$u_\rho = \frac{(S_\theta - \lambda S_\rho)\,\rho}{E} \quad \ldots\ldots\ldots\ldots (79)$$

Substituting the expressions for S_ρ and S_θ as given in Equations 72 and 73

$$u_\rho = \frac{r_1}{(1-k^2)\,E}\Big[(1+\lambda)\,(q_1 - q_2)\left(\frac{r_1}{\rho}\right) + (1-\lambda)\,(k^2 q_1 - q_2)\left(\frac{\rho}{r_1}\right)\Big] \ldots\ldots (80)$$

as the general equation for strain in a hollow cylinder under pressure.

Two mating cylinders are illustrated in *Fig.* 35 with common radius r. The dotted lines indicate the original position of the inner cylinder, and the dash lines the outer. Due to the interference between the two members, the outer cylinder will expand some amount u_b at the radius r, and the inner one will contract some amount u_a. Letting q equal the common pressure at the surface of contact and the sub-

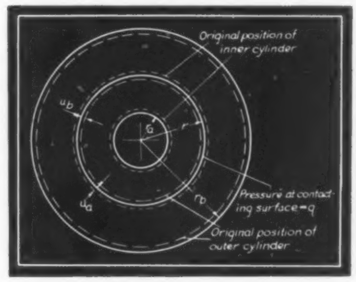


Fig. 35—Strain produced by the interference in a force fit. Total radial interference is the algebraic sum of u_a, u_b

script a and b distinguish between the inner and outer cylinders, respectively, and presuming first no loading on the inner boundary r_a or on the outer boundary r_b, substitution in Equation 80 gives for u_a and u_b

$$u_a = \frac{-qr}{E_a}(K_a - \lambda_a)$$

$$u_b = \frac{qr}{E_b}(K_b + \lambda_b)$$

The total radial interference $u = u_b - u_a$, from which, solving for q

$$q = \frac{\epsilon\, E_a\, E_b}{(K_a - \lambda_a)\,E_b + (K_b + \lambda_b)\,E_a} \ldots\ldots (81)$$

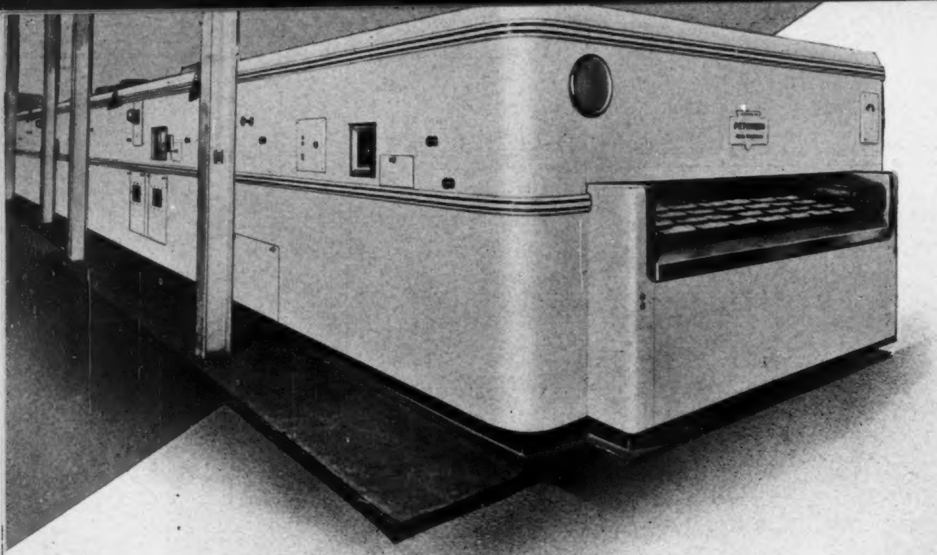
where $\epsilon = u/r$ is the "diametral interference per inch of diameter." The values of K_a and K_b may be taken from *Fig.* 32. With q evaluated, the stresses in the two cylinders can be determined by Equations 72 to 76. Change in the position of the four boundaries is determinable by means of Equation 80.

Result Applied to Bushing in Hub

The values of λ for various materials are not well determined. Fortunately, however, the error in λ has a relatively small effect on q. In the extreme case an error of 10 per cent will only affect q approximately 2 per cent. The usual case will be very much less than this.

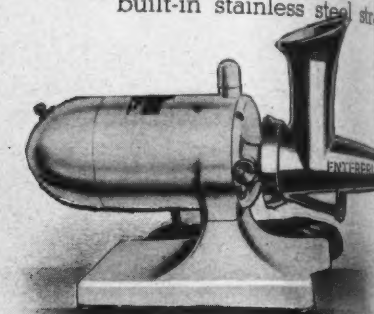
As an example of cylinders of different materials, a particular case of a bronze bushing pressed into a steel hub will be studied. The usual fit of .001-inch per inch of diameter will be assumed, that is $\epsilon = .001$. The effective radius of the hub will be taken at $2r$. Elastic constants for this bearing bronze will be assumed as $E_a = 14 \times 10^6$ and $\lambda_a = .35$. A ¼-inch thick bushing pressed into a 4-inch diameter hole will be considered. Substituting

(*Continued on Page* 110)



Petersen grill hearth traveling oven (above) is capable of controlling baking time of from 5 to 60 minutes. The traveling hearth is an endless grill belt permitting uniform heat distribution which, combined with a circulating system, assures accurate temperature control throughout the length and breadth of the oven. Accuracy and flexibility of heat control contributes much to the realized savings in fuel and power

Powered by a 115-volt unive[...] the Enterprise vegetable juic[...] (below) utilizes the pressure [...] juice extraction. The material [...] toward the small end of a tapere[...] extracted juices are delivered [...] built-in stainless steel str[...]



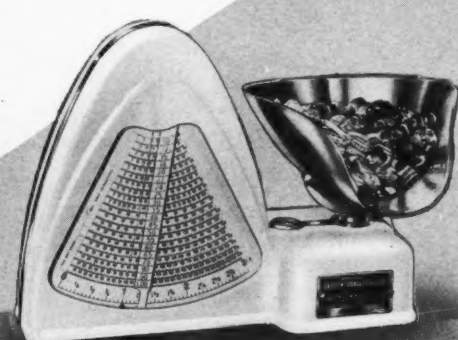
Electr[...] autom[...] penc[...] pen [...] of D[...] self-b[...] start[...] weig[...] top[...]

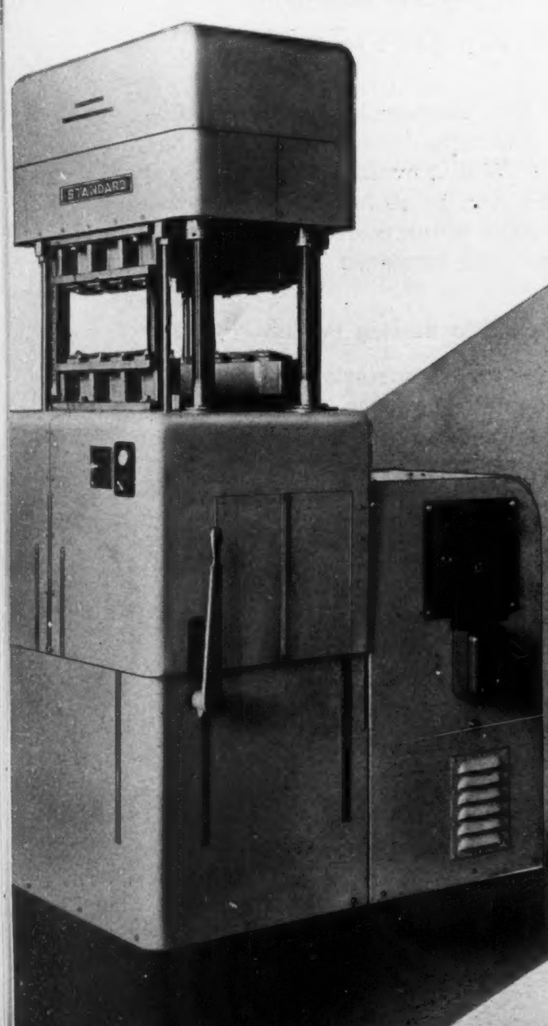
DesFeat[...] IN NE[...]ACH[...]

For [...] see page 1[...]

THIS MONT[...]ER. A[...] for plastics occasion[...] metal scarci[...] by the Hydraulic P[...] in develop[...] injection molding [...] culating flu[...] assures uniform he[...] plastic with[...] nation of flow mark[...] ation. Eq[...] piston type pressure [...] the press ca[...] pressure of 200 ton[...] sure in the[...] of 30[...] per square [...]

Semiautomatic Stokes molding press (below) incorporates a "slow closing" feature which minimizes rejects. As the press closes at normal high speed, an adjustable cam on the moving press element closes the rapid traverse valve. The hydraulic fluid then flows through a by-pass line. Throttle valve in this line controls speed of press closing. Opening of the press is controlled by rapid traverse valve. Speed adjustments can be made while press is in operation





Under the modern styling of the stainless steel-trimmed Hobart computing scale (above) is an accurate weighing mechanism incorporating jewel bearings, hydraulic shock absorbers and a thermostatic quick-stop device which provides automatic regulation of the indicator movement. Scale has a capacity of 3 pounds although it also is available with a 20-ounce range

Capable of automatically making metal master plates, wax master stencils and hectograph master copies from printed, typed or written material, the business machine (right) is made by Western Union. As the cylinder is rotated by an electric motor, a carriage moves laterally along the cylinder. Beam of light scans the original at from 100 to 120 lines per inch. A photocell receives the light or dark impulses and operates a solenoid on the duplicating end of the cylinder. Dark marks on the master cause a stylus operated by the solenoid to duplicate the copy

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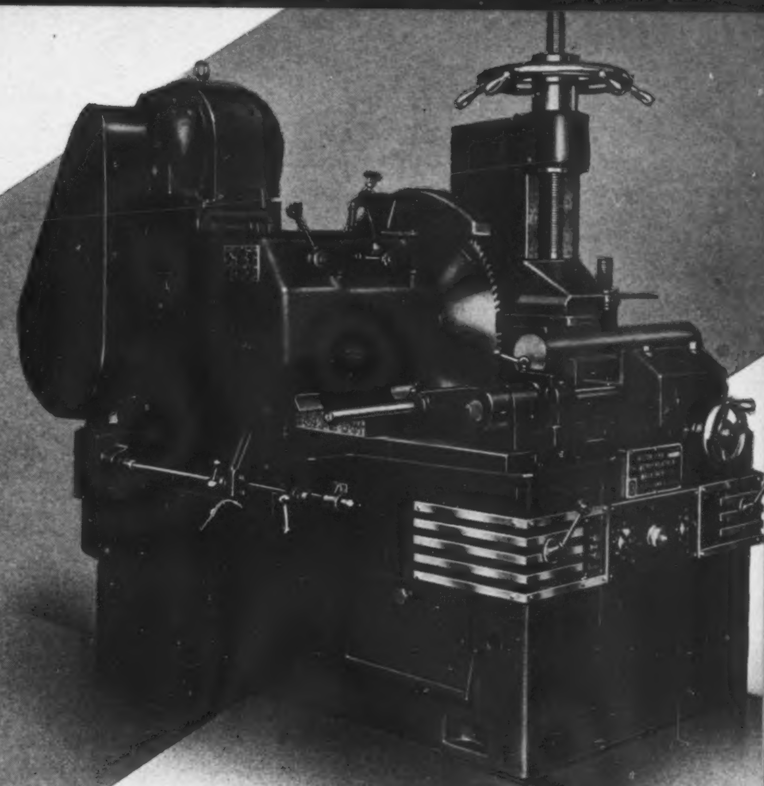
Electro-Pointer (below) sharpens pencils automatically and at high speed. The pencil is "dipped" into the sharpener as a pen is dipped into an inkwell. Molded of Durez, the device is powered by a self-balancing universal motor which is started by the pencil insertion. Well weighted, the unit rests firmly on table top or desk requiring no fastening

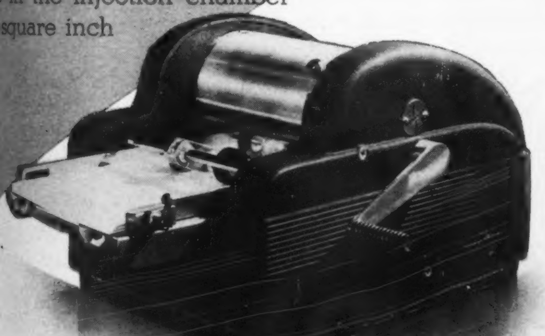


...es Features
...E[M]ACHINES

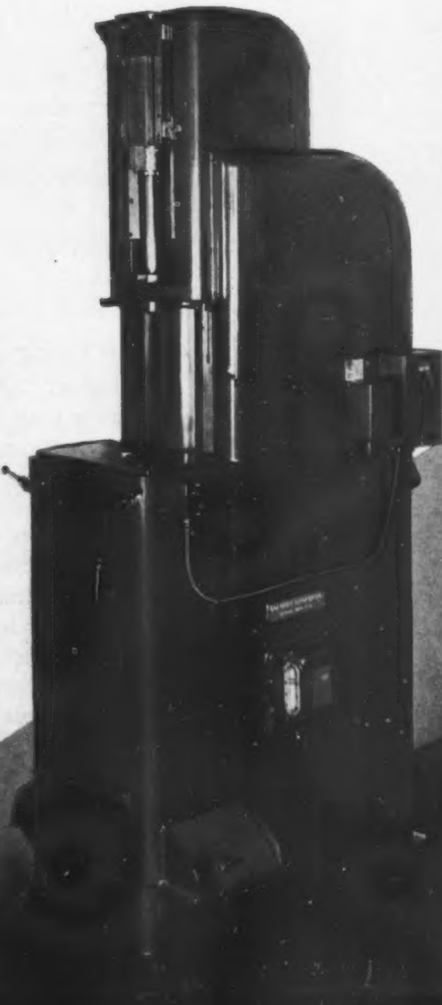
(For ... see page 124)

...N[T]ER. Accelerated demand ...sion ...etal scarcities was anticipated ...ic P... in developing this 12-ounce ...ng ...culating fluid heating system ... he... plastic with consequent elimi- ...mark... ation. Equipped with radial ...sure... the press can exert a clamping ... ton... sure in the injection chamber ... 30... per square inch

Hydraulic feed, cold sawing machine (above) designed and built by Motch & Merryweather, has all controls located on the front panel. Nine saw blade speeds may be selected ranging from 18 to 134 feet per minute. Hydraulic system provides power for feeding and clamping. Heat treated alloy steel spur and helical gearing assures minimum power loss. Drive gears are slidably mounted on splined shafts to facilitate speed changing



New liquid Universal duplicating machine (above) has no wick. A special roller has been designed which positively moistens the copy sheet without flooding regardless of speed of operation. Machine frame is cast aluminum for portability. Styled by Brooks Stevens, color scheme is black with chrome trim and red plastic operating handles. Ball or oilless bearings are used in all critical mechanical parts



Pull-down type Colonial broaching machines (right) have capacities of from 3 to 20 tons and strokes from 24 to 60 inches. Control of the tool is effected by two hydraulic cylinders; one is located in the top of the machine, the other in the base. Upper cylinder starts the pilot end of the broach through the part where it is gripped and drawn through by a sleeve type automatic gripper on the piston rod of the lower cylinder. Action is reversed for return stroke

MACHINE *Editorial* DESIGN

Cancellation of Expositions and Meetings Would Be Backward Step

QUERIES constantly are arising as to the advisability of continuing to hold engineering meetings and exhibitions. Fears are being expressed that due to defense preparations engineers will be unable to spare the necessary time away from their boards or desks and that consequent scarcity of attendance would limit the value of the gatherings.

No better evidence could be offered in rebuttal than the success of the recently held Machine and Tool Progress exposition sponsored by the A.S.T.E. In all phases—number of exhibitors, attendance at technical sessions held in conjunction, and attendance at the exposition itself—the meeting was more successful by far than any of those held in preceding years.

The same may be said of other engineering meetings, particularly when these are specifically geared to the needs of industry under the present program. At no time in the past has speed-up in defense been more imperative, leaving the field wide open for alert engineering organizations to sponsor meetings of abundant value.

It is to be hoped that, regardless of the pressure on engineers' time, attendance at such meetings will not be restricted. The value of interchange of ideas and discussion with fellow engineers, of the opportunity for consideration of information and material presented by authorities and displayed in exhibits, and the possibility of returning to the office with practical ideas having immediate applications in one's work, cannot be overestimated.

Design Data in Brief

TOP-RANK engineers are not necessarily those who carry in their heads the greatest fund of mechanical knowledge. Often the criterion applied is whether a man has the ability to locate the information needed at the essential time—assuming, of course, he has sufficient basic knowledge to enable him satisfactorily to apply the results of his findings.

It is in the endeavor to assist this type of resourceful engineer, constituting the majority of the country's machine designers, that a new department, "Engineering Data Sheets" is instituted in this issue. Many readers undoubtedly will find the information presented in these sheets of immediate value, and it is believed that others will wish to file the sheets for future reference.

"Data sheet" material suitable for filing has, since MACHINE DESIGN'S inception, been included in its technical articles, and this will continue to be done. Under the stress of current conditions the new supplementary data sheets, covering subjects of diversified interest, will serve to fill an increasingly pressing need.

Professional Viewpoints

MACHINE DESIGN welcomes comments from readers on subjects of interest to designers. Payment will be made for letters and comments published

" . . . counsel is valuable"

To the Editor:

Your March editorial entitled "Let's Plan Now To Overcome Shortage of Engineers" leads me to wonder whether or not the average practicing engineer appreciates what a potent influence he may wield in guiding potential engineers into their profession. Indeed, his help in guiding away from the engineering field those young men who are inherently unfitted for such work will be equally valuable to our country at this time when it is essential that everyone find his proper niche so that he may contribute his maximum efficiency.

Personnel workers in engineering schools are confronted daily by misfit students who firmly believe, for instance, that an ability and an urge to make model airplanes are the sole prerequisites for a successful career as an aeronautical engineer. The counsel of one practicing engineer carries as much weight with a student as the advice of a dozen personnel men, vocational advisers or faculty counselors.

We need craftsmen who know how to use their hands and we need them badly. We need engineers even more so, but they must be able to use their heads as well as their hands. Therefore, let every engineer take upon himself the responsibility of encouraging engineering talent into this profession and nonengineering talent into that particular field of work to which it is best suited individually.

There is no shortage of engineering minds in our country, but there certainly has been a marked scarcity of sound vocational guidance.

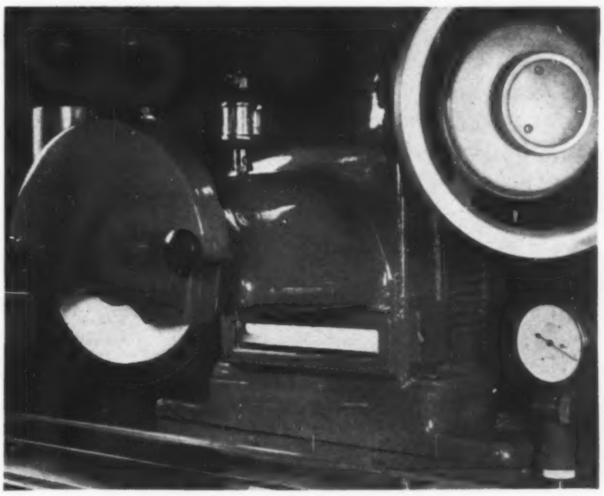
—H. A. BOLZ, *Asst. Prof. of Mech. Engrg.*
Purdue University

" . . . lights proper spot"

To the Editor:

An interesting application of built-in fluorescent lights, as discussed in Mr. Gaetjens article "How to Design for Adequate Built-in Lighting," is used on the Doall surface grinder and is shown in the accompanying photograph.

Among the features of this particular application is the fact that the light is flush-mounted, yet supplies illumination in exactly the correct spot for surface grinding operations. The light being fluorescent, is of the cold type, and consequently does



not heat the work on the table or annoy the operator. It is sealed into the housing of the machine to prevent any grinding or coolant from short-circuiting the contacts. The light is transmitted through a shatterproof glass window. Ballast and starter are located remote from the light, being mounted in an electrical control box in the base of the machine.

—J. W. WILKIE, *Vice President*
Continental Machines Inc.

" . . . when time is critical"

To the Editor:

The trend toward increased electrification of machine tools gives increased importance to the standardization of the symbols used in wiring

diagrams, as discussed in Mr. Elberty's article in your March issue. Mr. Elberty is rendering a signal service to all machine tool builders and users in crystallizing our ideas on this increasingly important subject.

While standards must occasionally change to meet changing circumstances, any standard is better than none. I am sure all users of machine tools will welcome clear diagrams, with supplementary "cross-the-line" diagrams when several electrical functions are involved and interlocking circuits are employed; and all of them made up of symbols that are familiar to engineers and therefore easily understood.

Every step toward simplification assumes new importance at a period like this when time is of critical importance and maximum production is our chief objective.

—TELL BERNA, *General Manager*
National Machine Tool Builders' Association

" . . . symbols are not consistent"

To the Editor:

Mr. Elberty's article in your March issue stresses the importance of wiring diagrams. If similar drawings as well as proposals for location of conduits, junction boxes, etc. were also made, much guessing could be eliminated. The resulting diagrams would be invaluable to the machine builder and ultimate customer.

In the symbols shown there appear to be several inconsistencies. For instance, power conductors should be shown in heavy lines for the main circuit contacts, oil circuit breakers, motor leads and series coils in *Figs.* 3 and 5. Also in *Figs.* 2 and 6, the differentiation between main and control circuits is not apparent.

—LEONARD JUDD
Westinghouse E. & M. Co.

M. D. regrets that the heavier thickness lines indicating power lines was not carried out consistently in all the drawings. It is hoped that no confusion will arise from the discrepancies.—ED.

" . . . leave nothing to chance"

To the Editor:

In connection with the design of lightweight motors for aircraft application as discussed in Mr. Fromm's article in your March issue, there are several items to which I would like to call attention.

While the design problems in connection with the building of lightweight, high-performance motors for aircraft are simplified in that the motors only operate for a few seconds and are then off for a long enough time to permit cooling, the problems are made more severe by the fact that absolute dependability is required in this particular type of service.

Though it is permissible to use considerably higher amperages than would ordinarily be used in a given size of motor, the designer must be sure that the brushes are satisfactory for these high amperages and will not give trouble in service by sticking or excessive arcing. Also, the mechanical construction of the motor must be such that it will stand abuse without mechanical failure in case any other parts of the electrical or mechanical system should fail to function, such as failure of a limit switch to operate—resulting in jamming of the mechanism at the end of the travel.

In such a case, the mechanical parts of the motor and its gearing must be capable of standing the impact blow without any failure. The most important thing to keep in mind is that if there is any conceivable thing in the motor or the mechanism that can possibly give trouble it is bound to do so at some time during the life of the motor and, therefore, this possibility should be eliminated to begin with rather than after trouble has been experienced. Nothing must be left to chance on the theory that it will probably turn out to be all right.

—T.A. WELLS, *Vice Pres. & Chief Engineer*
Beech Aircraft Corp.

To the Editor:

Mr. Wells in his letter commenting on my article on aircraft motors has made a good point in stressing dependability. Where life itself may depend upon the satisfactory operation of an electric motor, dependability cannot be over-emphasized.

—W. H. FROMM, *Aviation Division*
Dumore Company

" . . . is suitable matrix alloy"

To the Editor:

Use of matrix metals for locating bearings and round ways in unit cast frames as discussed in your article "Shell Lathe Design Unfettered by 'Standard Practice' " is interesting. About 3400 machines, however, utilized this construction successfully during the last war and 68 per cent of all shells six inches and larger that were made in the U. S. and Canada between the years 1915 and 1918 were made on these machines.

A suitable alloy is a mixture of lead and antimony containing 1 per cent more antimony than the eutectic alloy—which is 87 lead and 13 antimony. This alloy (86 lead, 14 antimony) has a melting point of 489 degrees Fahrenheit and should be poured at 550 degrees Fahrenheit—within ten degrees maximum either way. It has an ultimate compression strength of 4000 pounds per square inch.

—LUCIEN I. YEOMANS
Chicago

Mathematical Solution of 4-Bar Linkages

IN inaugurating Engineering Data Sheets, MACHINE DESIGN'S primary objectives are: (1) Simplification of mechanical calculation and analysis, and (2) concise and readily available presentation of engineering information.

Guy J. Talbourdet's work, "Mathematical Solution of 4-Bar Linkages", introduces this department. Engineers who have carried out the laborious graphical analyses of linkage systems far less complicated than that illustrated in Fig. 1 will appreciate the unprecedented accuracy possible by this analytical method. Furthermore, many linkage combinations which are virtually impossible of solution by the graphical technique are readily solvable by the application of Mr. Talbourdet's formulas, his preface to which follows

DESPITE the universal occurrence of the 4-bar linkage or quadric crank in machines there have been, heretofore, no consistently accurate methods for determining the kinetics of its motion. All problems of this type have been solved by the graphical method of relative displacements, velocities and accelerations. Although this method suffices for many simple linkage systems it requires careful draftsmanship to obtain accurate results. In the design of precise, high speed machines, the graphical method has definite limitations.

Presentation of the mathematical approach to these problems will be made in four parts as follows:

PART 1: Angular displacement, velocity and acceleration of the driven crank as a function of both variable and uniform motion of the driving crank.

PART 2: Angular displacement, velocity and acceleration of any driven crank in a series of 4-bar linkages as a function of the motion of any driving crank in the series.

PART 3: Motion of any point on the connecting link of a 4-bar linkage as functions of a uniform angular motion of the driving crank.

PART 4: Balancing of linkage systems.

Parts 1 and 2 of the series appear on the fol-

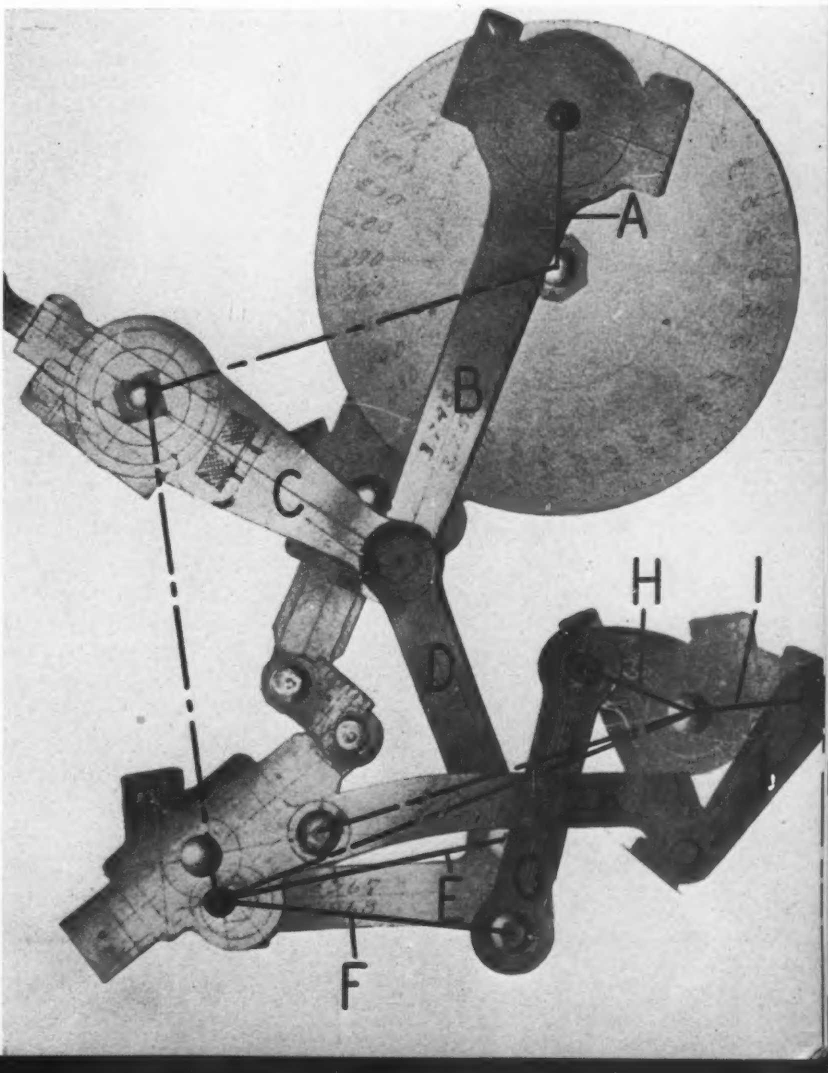


Fig. 1—Right—Systems of four 4-bar linkages in series similar to that illustrated lend themselves readily to solution by the mathematical technique

lowing pages. Parts 3 and 4 will be presented in successive issues.

Although the equations may, at first sight, seem too lengthy to be practical, a closer examination shows that many terms are common and are often repeated. This fact combined with the increasing use of calculating machines in engineering work makes the computations a routine procedure.

As an illustration of the practical application of this method, the following example is given:

A machine designed to operate at 900 revolutions per minute was constructed with a mechanism consisting of four 4-bar linkages connected in series. Tests on this machine were not entirely satisfactory because of excessive noise in operation and high power requirements. Therefore, a cardboard model of the mechanism, shown in *Fig.* 1, was made for analysis.

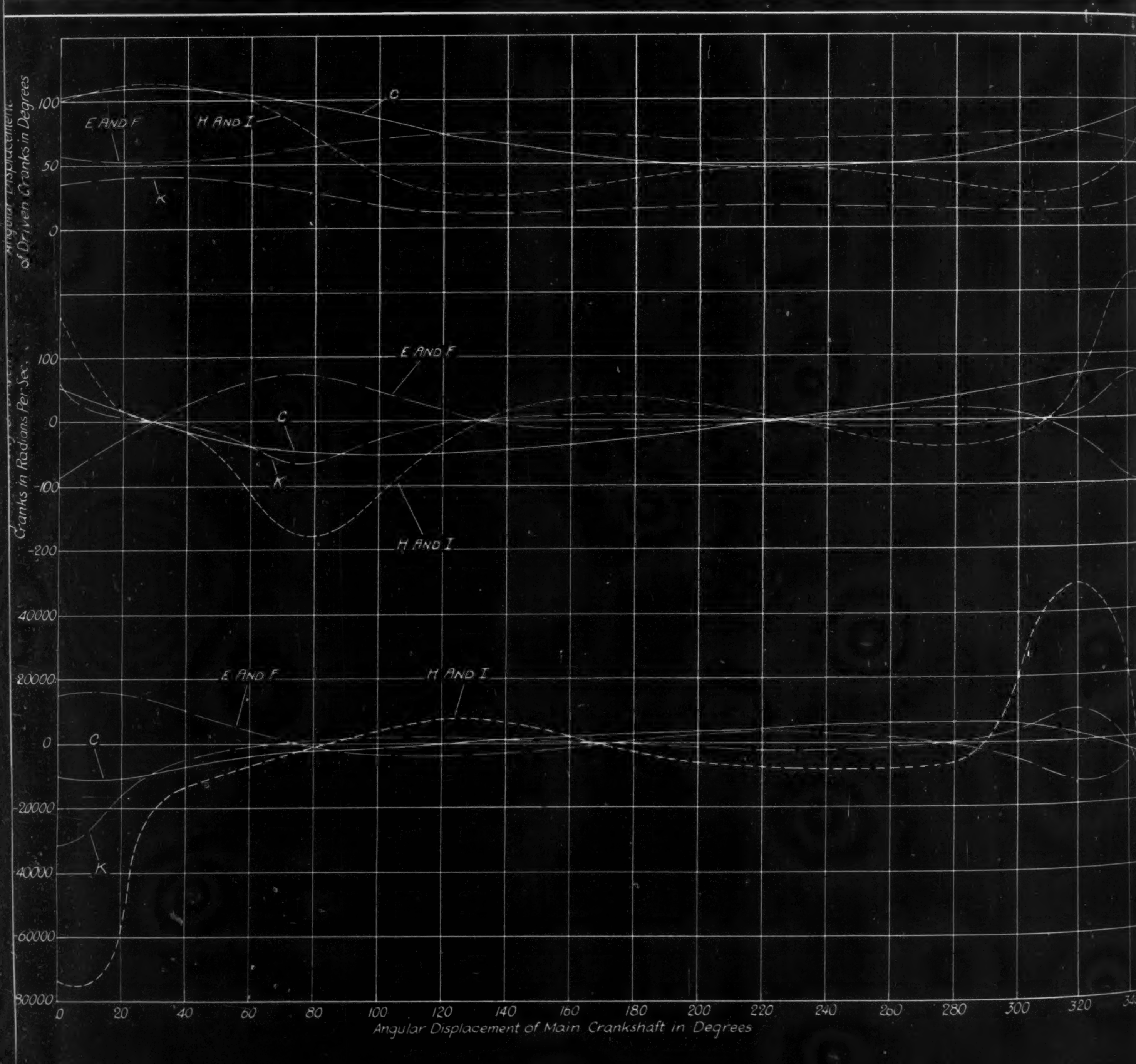
Mathematical investigation of the linkage system showed the results in *Fig.* 2 (below). Examination of this graph readily showed that the angular accelera-

Four 4-Bar Linkages in Model

Linkage Number	Driving Crank	Connecting Link	Driven Crank
1	A	B	C
2	C	D	E
3	F	G	H
4	I	J	K

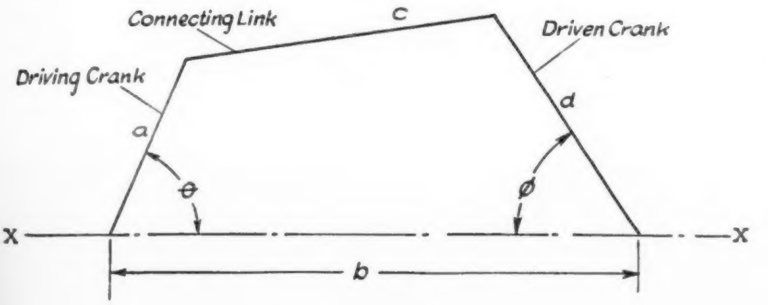
tion of the driven member *H* in the third 4-bar linkage was high enough to make the mass inertia of that member and the fourth driving crank *I* (*Fig.* 1) an important factor in the smoothness with which the mechanism operated.

On the basis of a force analysis which followed, the inertias of these members were reduced to a minimum. A torsional deflection in the shaft connecting the third driven crank *I* to the fourth driving crank *K* was also introduced by reducing the diameter of the shaft. As a result of these changes, the power required to operate the machine decreased from 1.6 horsepower to 1.4 horsepower at 600 revolutions per minute and from 2.75 to 2.08 horsepower at 900 revolutions per minute. Noise of operation was also materially reduced.



Part I—Analysis of Single 4-Bar Linkage

By Guy J. Talbourdet
Research Division
United Shoe Machinery Corporation



Calculate the values of the following factors from the geometry of the linkage:

$$K = a^2 + b^2 - c^2 + d^2$$
$$A = a \operatorname{Sin} \theta$$
$$B = a^2 + b^2 - 2ab \operatorname{Cos} \theta$$
$$D = K - 2ab \operatorname{Cos} \theta$$
$$S = \sqrt{4d^2B - D^2}$$

θ = Angular displacement of driving crank.

ϕ = Angular displacement of driven crank.

$\frac{d\theta}{dt}$ and $\frac{d^2\theta}{dt^2}$ = Angular velocity and acceleration respectively of the driving crank.

$\frac{d\phi}{dt}$ and $\frac{d^2\phi}{dt^2}$ = Angular velocity and acceleration respectively of the driven crank.

Then:

$$\phi = Tan^{-1}\frac{A}{b - a \operatorname{Cos}\theta} + Cos^{-1}\frac{D}{2d\sqrt{B}}$$

And:

$$\frac{d\phi}{dt} = \frac{d\theta}{dt}\left[\frac{a}{B}(b \operatorname{Cos}\theta - a) - \frac{Ab}{S}\left(2 - \frac{D}{B}\right)\right]$$

And:

$$\frac{d^2\phi}{dt^2} = \frac{d^2\theta}{dt^2}\left[\frac{a}{B}(b \operatorname{Cos}\theta - a) - \frac{Ab}{S}\left(2 - \frac{D}{B}\right)\right]$$
$$+ \left(\frac{d\theta}{dt}\right)^2\left[\frac{2A^2b^2}{BS}\left(1 - \frac{D}{B}\right) + \left(2 - \frac{D}{B}\right)\left(\frac{2A^2b^2(2d^2 - D)}{S^3} - \frac{ab \operatorname{Cos}\theta}{S}\right)\right.$$
$$\left. - \frac{Ab}{B}\left(1 + \frac{2a(b \operatorname{Cos}\theta - a)}{B}\right)\right]$$

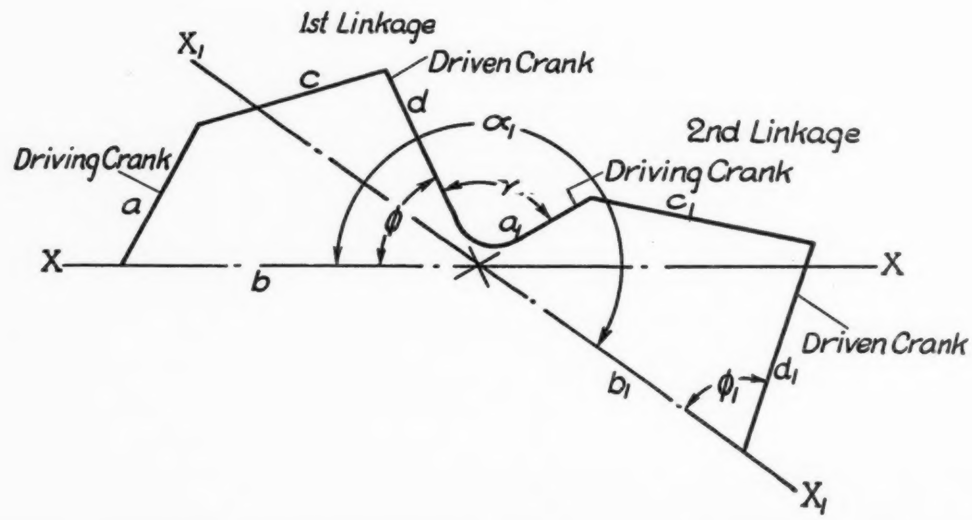
When the motion of the driving crank is uniform, $\frac{d\theta}{dt} = \omega$, and $\frac{d^2\theta}{dt^2} = 0$, in which case we have:

$$\frac{d\phi}{dt} = \omega\left[\frac{a}{b}(b \operatorname{Cos}\theta - a) - \frac{Ab}{S}\left(2 - \frac{D}{B}\right)\right]$$

And:

$$\frac{d^2\phi}{dt^2} = \omega^2\left[\frac{2A^2b^2}{BS}\left(1 - \frac{D}{B}\right) + \left(2 - \frac{D}{B}\right)\left(\frac{2A^2b^2(2d^2 - D)}{S^3} - \frac{ab \operatorname{Cos}\theta}{S}\right)\right.$$
$$\left. - \frac{Ab}{B}\left(1 + \frac{2a(b \operatorname{Cos}\theta - a)}{B}\right)\right]$$

Part II—Analysis of 4-Bar Linkages in Series



When a 4-bar linkage is connected to another as for instance by a bell-crank as shown in the illustration, the angular motion of the driven crank of the second linkage may be determined as a function of that motion of the driven crank of the first linkage. Apparently, for more than two linkages in series, a repetitive application of the same procedure will result in the determination of the motion of any crank in the series.

$\frac{d\phi}{dt}$ = Angular velocity of first driven crank.

$\frac{d^2\phi}{dt^2}$ = Angular acceleration of first driven crank.

$\frac{d\phi_1}{dt}$ = Angular velocity of second driven crank.

$\frac{d^2\phi_1}{dt^2}$ = Angular acceleration of second driven crank.

For simplicity let $\gamma = 0$.

Then

$\alpha_1 - \phi$ = Angle between driving crank of second 4-bar linkage and $X_1 - X_1$ axis. If γ does not equal 0, this angle is $\alpha_1 - \phi - \gamma$. Calculate the values of the following factors from the geometry of the linkage.

$$K_1 = a_1^2 + b_1^2 - c_1^2 + d_1^2$$
$$A_1 = a_1 \, Sin \, (\alpha_1 - \phi)$$
$$B_1 = a_1^2 + b_1^2 - 2a_1b_1 \, Cos \, (\alpha_1 - \phi)$$
$$D_1 = K_1 - 2\, a_1b_1 \, Cos \, (\alpha_1 - \phi)$$
$$E_1 = a_1b_1 \, Cos \, (\alpha_1 - \phi)$$
$$H_1 = a_1b_1 \, Sin \, (\alpha_1 - \phi)$$
$$S_1 = \sqrt{4d_1^2 B_1 - D_1^2}$$

Then

$$\phi_1 = Tan^{-1} \frac{A_1}{b_1 - a_1 \, Cos \, (\alpha_1 - \phi)} + Cos^{-1} \frac{D_1}{2\, d_1 \sqrt{B}}$$

Since α_1 is constant

$$\frac{d\phi_1}{dt} = \frac{d\phi}{dt}\left[\frac{a_1^2 - E_1}{B_1} + \frac{H_1}{S_1}\left(2 - \frac{D_1}{B_1}\right)\right]$$

And

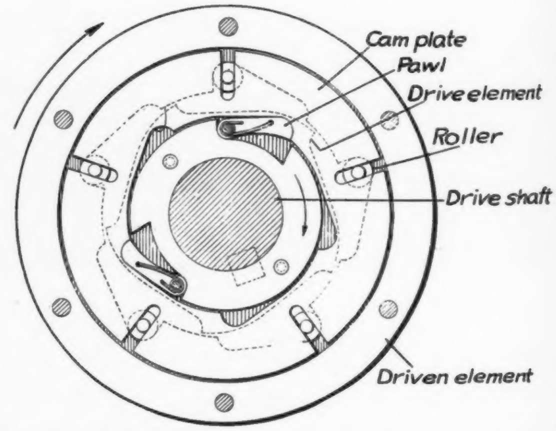
$$\frac{d^2\phi_1}{dt^3} = \frac{d^2\phi}{dt^2}\left[\frac{a_1^2 - E_1}{B_1} + \frac{H_1}{S_1}\left(2 - \frac{D_1}{B_1}\right)\right] + \left(\frac{d\phi}{dt}\right)^2\left[\frac{H_1}{B_1}\left(\frac{2\,(a_1^2 - E_1)}{B_1} - 1\right)\right.$$
$$\left. + \frac{2H_1^2}{S_1B_1}\left(1 - \frac{D_1}{B_1}\right) + \left(2 - \frac{D_1}{B_1}\right)\left(\frac{2H_1^2\,(2\,d_1^2 - D_1)}{S_1^3} - \frac{E_1}{S_1}\right)\right]$$

Noteworthy PATENTS

Provides Instantaneous Engagement

EFFECT of centrifugal force on the rollers of an overrunning clutch will delay the re-engagement of the drive for an appreciable time after the driven element slows down to the driver speed or the driving element speeds up to the driven speed. Conventionally, springs are used to overcome this effect. However, if the springs are too strong, the rollers are urged into contact with both the driven and driving elements even while the clutch is overrunning, resulting in excessive noise and wear. If the springs are too weak the result is the same as if no springs at all are used except that re-establishment of positive drive will occur at a slightly higher speed. In this patent, assigned to the Kinney Mfg. Co., positive re-engagement is assured as soon as the driving and driven elements approach the same speed, regardless of what that speed may be.

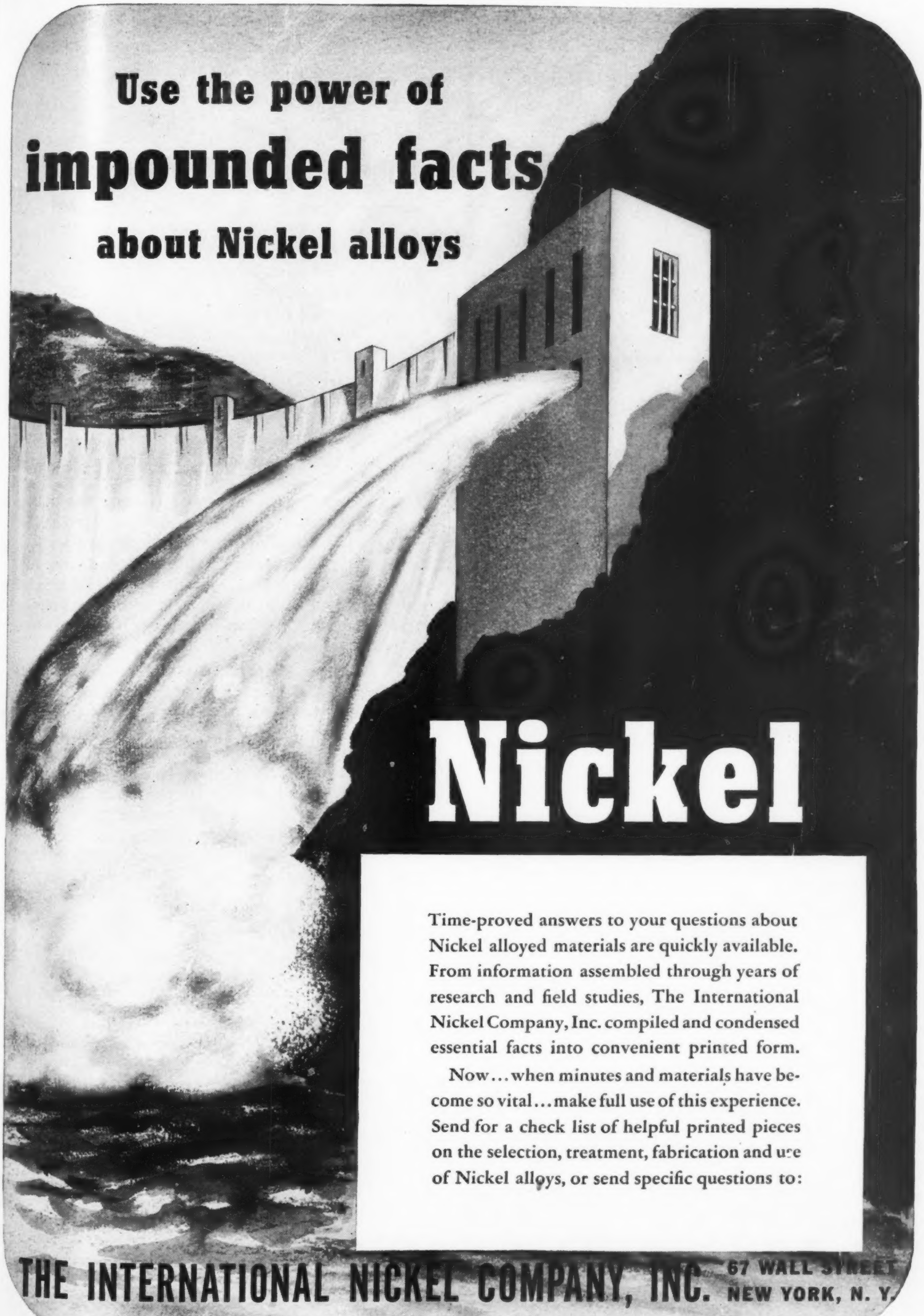
The device consists essentially of a conventional overruning clutch, with the exception that on either side of the clutch a thin cam plate is used which



Cam plate operated by pawls assures immediate overtake of driven member of overrunning clutch through positive roller engagement

constrains the ends of the rollers in radial slots. The hole in the cam plate has a contour adapted to be engaged by pawls carried by drive element.

Position of the parts of the clutch is illustrated at the time the speed between the driving and driven elements becomes the same. The pawls are in engagement with the cam plate, and it is evident that further rotation of the shaft will result in driving the cam plate clockwise relative to the driven element thereby bringing the load-carrying rollers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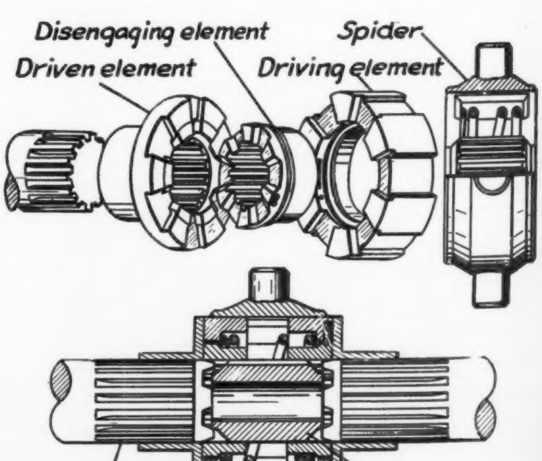
into driving engagement with the driven element. Thus, positive engagement is assured without in any way detracting from the overrunning properties of the clutch.

Uniform Torque Output Assured

DELIVERY of full torque to one or both driving wheels of an automobile is provided by this patent assigned to C. F. Gobright Inc. If one driving wheel is on ice or sand or otherwise loses its driving frictional engagement with the road, the conventional differential permits that wheel to spin and delivers no torque to the other wheel. While this condition could be remedied by the use of a pair of overrunning clutches instead of a differential, the vehicle would then be unable to reverse. The mechanism discussed herein provides all the advantages of the overrunning clutch for forward motion and also permits the vehicle to reverse.

A spider designed to be installed in the ring gear cage of a conventional differential contains, splined to its interior, a pair of driving elements which are thrust laterally outward by a spring. The upper view of the illustration shows the parts of the left side of the assembly; the right side is identical. Bearing against a shoulder inside each driving element is a disengaging element which has beveled teeth (mating with the disengaging element) on the inner portion. The two disengaging elements are splined to a sleeve.

In rounding a corner, the outer wheel rotates faster than the inner. The beveled cam teeth of the driven element of the outer wheel thrust the disengaging element, and hence the drive element, out of engagement. Due to the compression of the spring, this results in the other clutch being held even more firmly in engagement for supplying full torque to the slower wheel. It is apparent that even though one wheel has inadequate frictional contact with the road, the other will still drive in either forward or reverse direction.

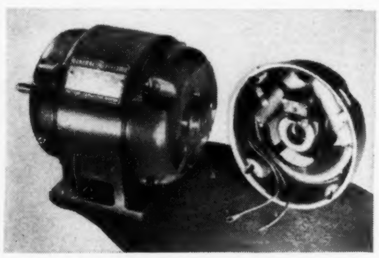


Positive clutches with automatic disengaging means provide delivery of full torque to the slower moving shaft or axle

New PARTS AND MATERIALS

High and Low Torque Motor Offered

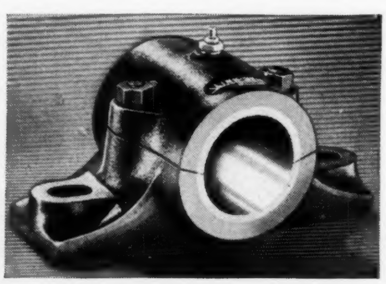
DESIGNED to meet a number of varied industrial applications, a new Tri-Clad capacitor motor has been added to General Electric Company's recently announced line of polyphase induction motors. The motor is available in two types, KC and KCJ, and may be obtained with either ball or sleeve bearing construction. Type KC is designed for applications requiring moderate starting torques while Type KCJ is for use where high starting torque is required. On the normal-torque motor the ca-



pacitors are inside the end shield and on the high-starting torque motor, 1½ horsepower and larger, the capacitors are mounted in a compact case on top of the motor frame. Better mechanical and electrical protection is provided as well as improved bearing design and lubricating arrangements. Transfer switch, consisting of centrifugal mechanism on the rotor shaft and stationary switch located in the end shield, is a simple device transferring the motor starting-to-running connection. As motor approaches full speed, centrifugal force moves the collar of rotating mechanism away from switch by snap action, causing the switch to open.

Bearing Housing is "Cleanlined"

ANNOUNCED by Link-Belt Co., 307 North Michigan avenue, Chicago, a new babbitted bearing with maroon-colored crackle-finish housing has been



designed for moderate speed and power requirements. It is available from stock in twenty-four sizes, for shafts of ½ to 3 inches in diameter. The bearing has compactness with proper metal distribution to assure maximum strength and rigidity within minimum overall dimensions. Sloping surface between cap and base maintain concentricity and relieve the cap bolts from direct strain of side thrust. The shims between base and cap may be removed for needed adjustment. Base is ground to close tolerances and is properly recessed to compensate for inaccuracies in surface of structural support. The cap in bearing is tapped for grease cup or hydraulic fitting and a lubrication groove in cap bore assures proper distribution of lubricant. To provide accurate bearing surfaces for collars, pulleys, gears or sprockets the bearing ends are finished.

Filter Uses Magnetized Screens

FUNDAMENTALLY different from conventional filters and separators, the permanent magnet "Ferro Filter", produced by S. G. Frantz Co. Inc., 161 Grand street, New York, is designed to remove iron particles from suspension in lubricating oil. It catches ferrous particles as fine as one micron (.001 millimeter). These if not removed can break oil film and bridge bearing clearances. Full flow and nonclogging filter located in the oil pipe line becomes part of the circulating oil system and safeguards the oil from harmful iron particles. The filter consists essentially of a stack of magnetized screens enclosed in a cylindrical casing through which oil flows. The screens have triangle mesh openings of about ⅛-inch and while they offer very little resistance to flow, their design presents hundreds of feet of strongly magnetized edges which comb and recomb the oil to insure efficiency of removal. The magnetic particles are held firmly to the screen edges until filter is cleaned.

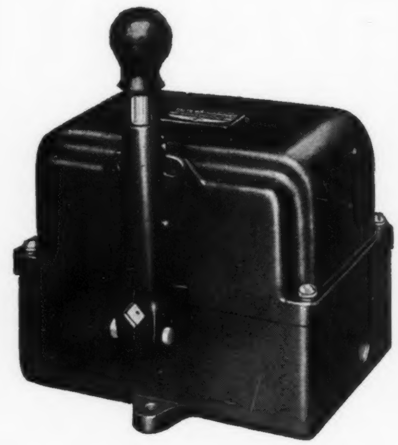
Controller Speeds up Production

THROUGH reduced fatigue, the new line of cam-operated, mill duty controllers introduced by Cutler-Hammer Inc., 315 North Twelfth street, Milwaukee, speed up production because of ease of operation. Positive feel of all speed positions is obtained by using an adjustable compression type of star wheel spring. The controllers are available

(Continued on Page 80)

(Continued from Page 74)

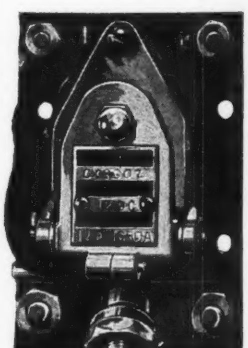
in two, three and multispeed types and have all of the following outstanding points: Contacts are vertical, double-break, silver to silver; camshaft operates on ball bearings sealed against dust; and an



easily accessible terminal board simplifies installation and service. A heavy cast case and cover, for either separate or bench board mounting, provide protection from dust and mechanical injury. Controllers have such optional features as spring return, off position latch, and two, three or five speeds.

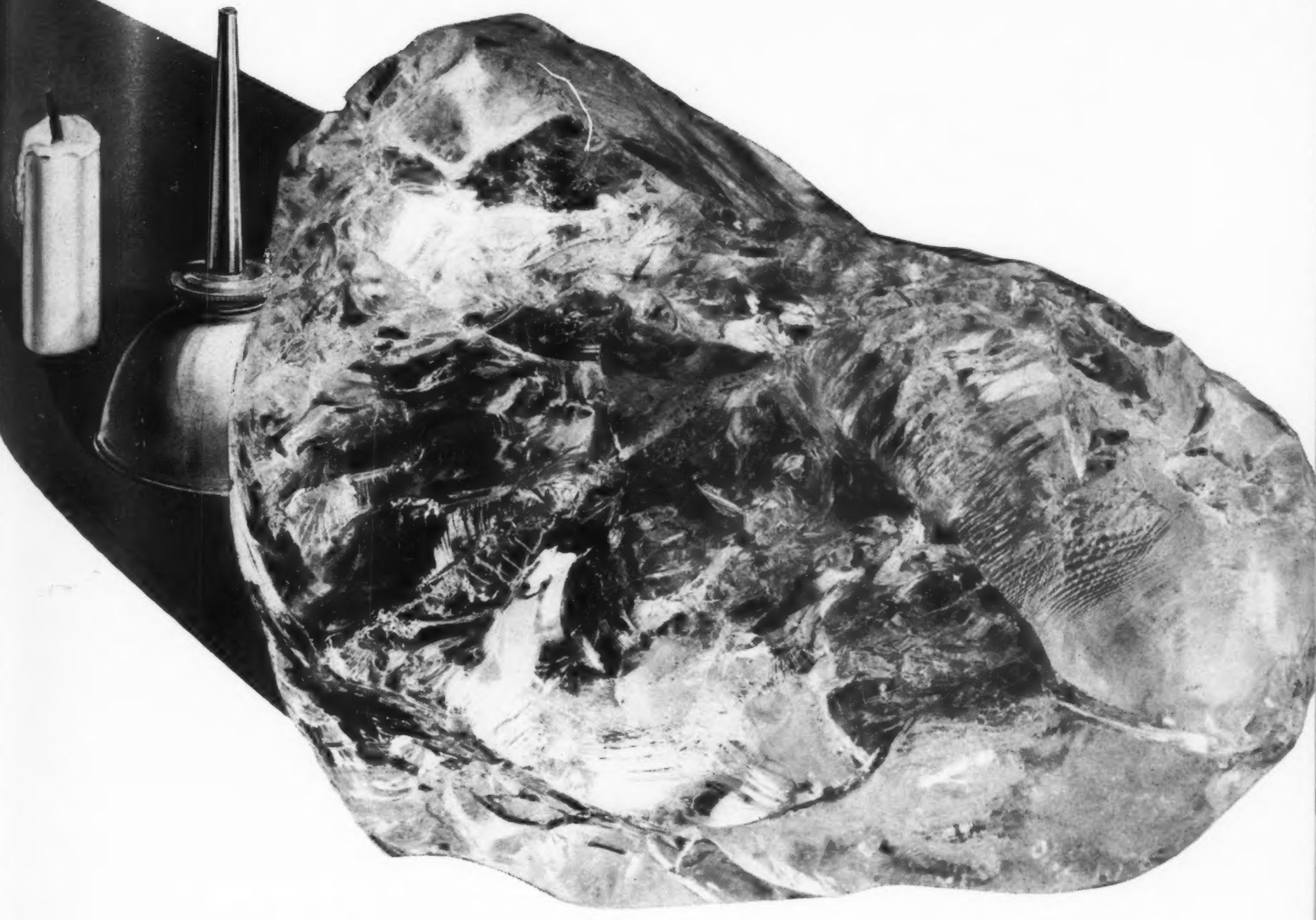
Relays Designed for High Loads

TWO new lightweight relays have been designed by Struthers Dunn Inc., 1335 Cherry street, Philadelphia. While primarily for aviation service for control of air flaps and similar applications where dependability is required, they are proving popular for industrial requirements. The standard type C3007 relay is single-pole, single-throw with normally open contacts rated 50 amperes at 12 or 24 volts direct-current, and will stand an inrush of 100 amperes. This relay, 2¾ inches high, 1⅞ inches wide, 2 inches deep, and weighing 8 ounces, has a contact pressure of 10 pounds. Type CX3007 relays are mounted on a common base for motor reversing and are 3⅝ inches high, 4 inches wide, 3 inches deep, and weigh 15 ounces. A larger relay, CXA3008, is rated 100 amperes at 12 volts direct-current, or 80 amperes at 24 volts direct-current, and will stand an inrush of 300 amperes. Contact pressure of this type is 15 pounds and the size is 3¾ inches x 2½ inches x 2⅝ inches. Weight is 1 pound. Each unit can be modified or adapted to meet individual requirements.



Synchronous Motor Time-Switch

SWITCH made with only two exposed gears, all others operating in a sealed oil bath and consequently always lubricated and free from dust, is



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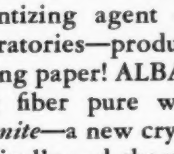

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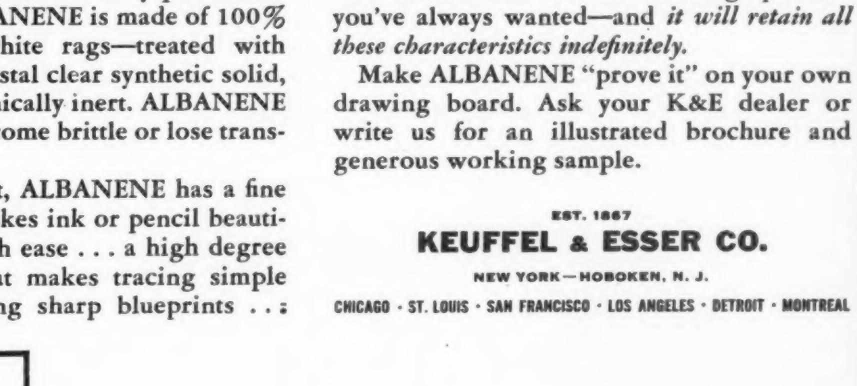
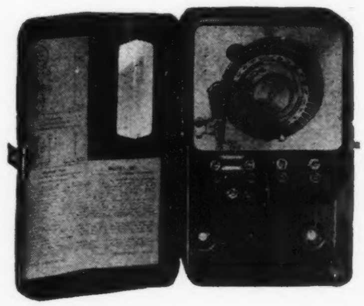


the result of engineering research at Paragon Electric Co., 37 West Van Buren street, Chicago. The operation of the time switches is accomplished by slow speed, industrial type, self-starting synchro-



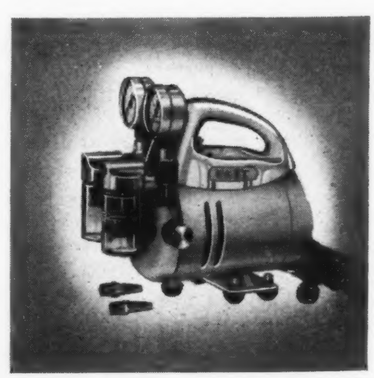
nous motor of 450 revolutions per minute. For actuating off-tripper in the switch, there is a reserve power of 40 pounds or an equivalent of 2000 per cent. It operates successfully at temperatures as low as 20 degrees below zero. The case is compact, made of 18-gage steel with a seamless drawn cover and base, streamlined and modern.

Extruded Plastics Line Expanded

CONTINUOUS extruded plastics are now being produced by the Plastics Division of Erie Resistor Corp., 640 West Twelfth street, Erie, Pa., as a result of its newly expanded molding facilities. Custom designed moldings in the form of solid rods, tubing, half rounds or any other shapes that lend themselves to extrusion, are molded in continuous lengths up to a maximum diameter of ⅞-inch. A wide range of materials in any desired color, from opaque to translucent, are available.

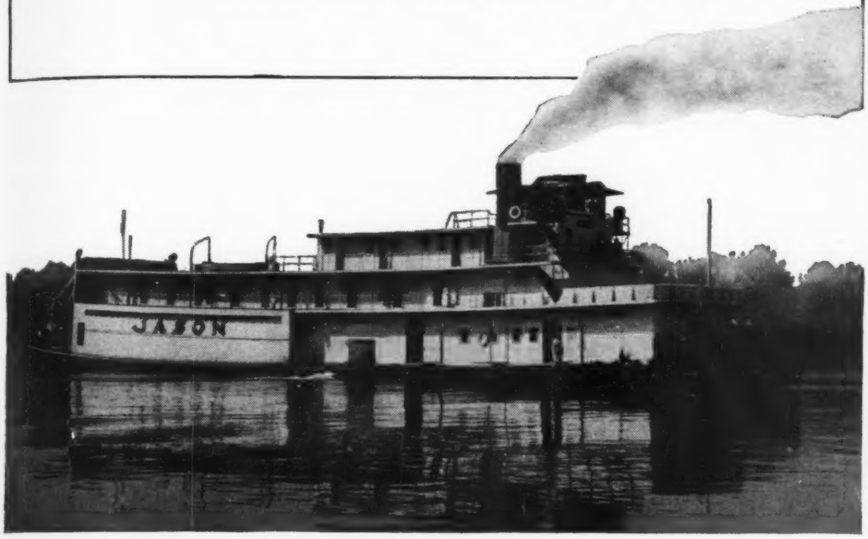
Pump Creates Vacuum or Pressure

FOR both vacuum and pressure operation, the air pump shown is a new model introduced by Gast Mfg. Corp., Hinkley street, Benton Harbor, Mich. Complete with a 1/12-horsepower motor, all parts necessary for vacuum and pressure operation are contained in the integral unit. The pump is



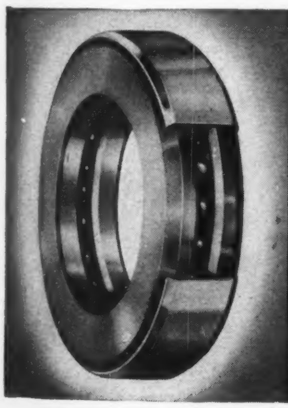
compact and light in weight, and in its deluxe model includes chromium-plated and polished parts for hospital, laboratory and surgical work. It may also be had in black metal finish for industrial use as

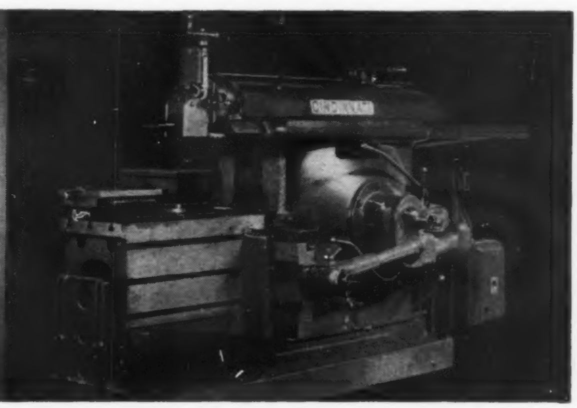
IN THE NEWS
WITH BANTAM BEARINGS



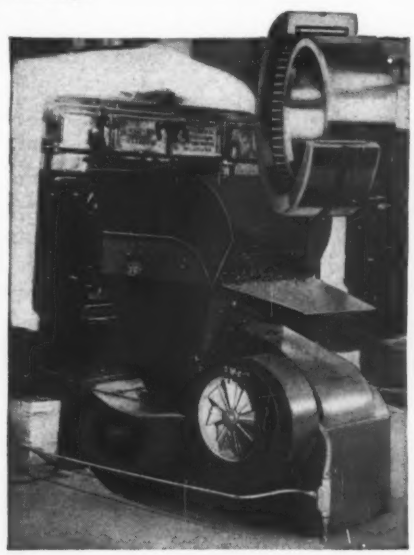
THE MISSISSIPPI STEAMBOAT of today combines the advances of up-to-the-minute engineering design with the long-recognized advantages of the paddle-wheel steamer for shallow river service. Among the newest of river boats is the *Jason*, designed and built by Marietta Manufacturing Company. In keeping with its progressive design, Bantam Quill Bearings are used on the cam rollers that actuate the valves of the *Jason's* engines.

FOR LONGER BEARING LIFE, Bantam employs bronze cages in many of its bearings, because the bronze cage surfaces are long-lasting, and provide a safeguard against wear on the steel rollers. Photo shows welding of the bronze cage for a Bantam Precision Tapered Roller Bearing.

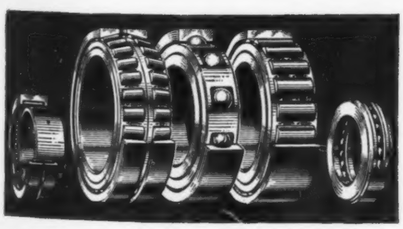




FAST, EFFICIENT PRODUCTION is the keynote of the comprehensive line of shapers built by The Cincinnati Shaper Co. The line includes 24 models in 10 sizes and 5 types—and *all* use Bantam Ball Thrust Bearings on the rail and table elevating mechanism, to take the entire thrust load when table is raised or lowered.



HEAT, GRIT, AND HEAVY LOADS are normal operating conditions for mechanical stokers, which must perform reliably even if neglected or overloaded. Combustion Engineering, Inc. assures long life and efficient operation of its Skelly Stoker Units by installing Bantam Quill Bearings at vital points. For further information on this compact, high-capacity anti-friction bearing, write for Bulletin B-104.



EVERY MAJOR TYPE of anti-friction bearing is included in Bantam's line—straight roller, tapered roller, needle, and ball. Bantam serves every industry with a wide range of standard bearings that meet many normal requirements. Bantam engineers offer unbiased advice on selection of standard bearings—and design custom-built bearings in large sizes or special types for unusual conditions. If you have an exceptionally difficult bearing problem, TURN TO BANTAM.

BANTAM BEARINGS

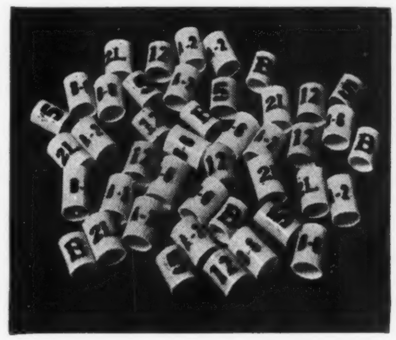
STRAIGHT ROLLER · TAPERED ROLLER · NEEDLE · BALL

BANTAM BEARINGS CORPORATION • SOUTH BEND • INDIANA

a machine accessory. Of rotary type, the pump operates without gears, springs or valves and is asserted to be efficient, vibrationless and quiet in performance. It can be converted by a slight adjustment from vacuum to pressure service. The pump and motor measure 8½ inches x 4½ inches and the air capacity is 0.5 cubic feet per minute.

Varnish-Impregnated Tubing

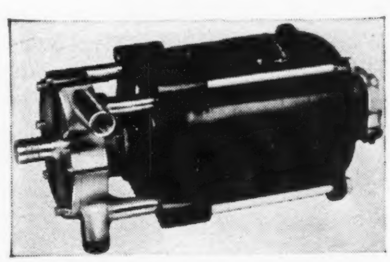
SUITABLE for aircraft engines and many other applications, an improved ignition marker has been perfected by Irvington Varnish & Insulator Co., 24 Argyle Terrace, Irvington, N. J. Consisting of lengths of inside and out, varnish-impregnated insulating tubing, on which suitable identification is screen-printed with a specially formulated ink,



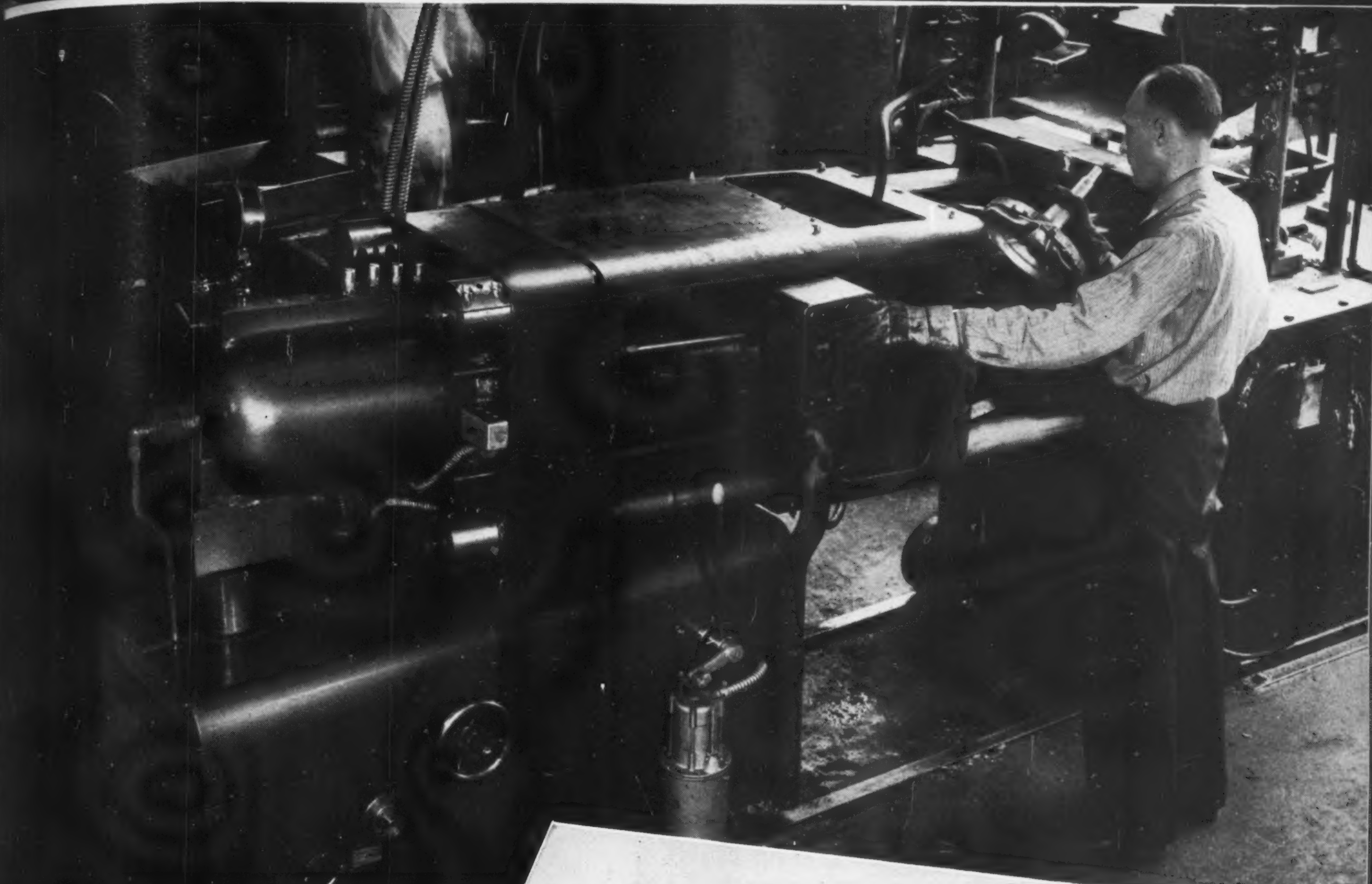
the marker is furnished in two standard lengths which are slipped over wires to be identified. Available in two diameters (nominal I.D. .263 or .294) with either of two heights of symbols, this marker is an improvement over the former type in which the identification numerals and letters were printed on cellophane strips which in turn were fastened to insulating tubing.

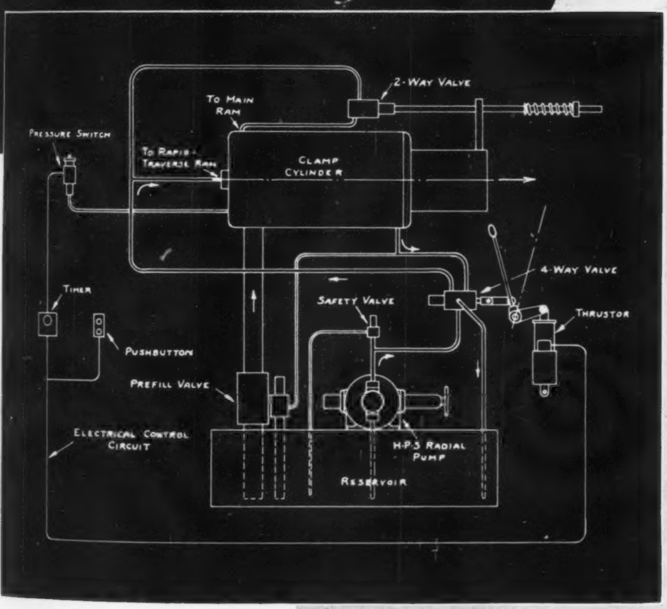
Pump Has Explosion-Proof Motor

MIDGET size pump with explosion-proof motor is announced by Eastern Engineering Co., 45 Fox street, New Haven, Conn. Compact in size and construction, this pump has a maximum capacity of 5.7 gallons per minute with a maximum



pressure of 16 pounds per square inch, and can be used in all industrial, pilot plants, laboratory and experimental applications where an explosion-proof unit is necessary. The pumps, furnished in stainless steel, monel metal, chromium-plated bronze, brass, cast iron, Hastelloy and other alloys, are 10 inches long, 5½ inches high and 5 inches wide; weight is 18 pounds. The motor is either 110 or 220 volts single phase, ½ horsepower at 3450 revolu-

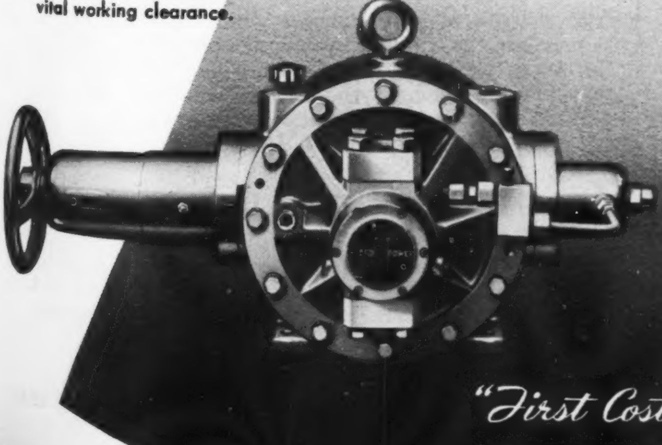




Continuous hydraulic machine performance depends primarily upon the pressure generator and its design. The HYDRO-POWER radial piston type pump is of exclusive design incorporating—

(1) Precision tapered Timken Roller Bearing mounting of rotor.

(2) Means for easy external adjustment of these rotor bearings to maintain vital working clearance.

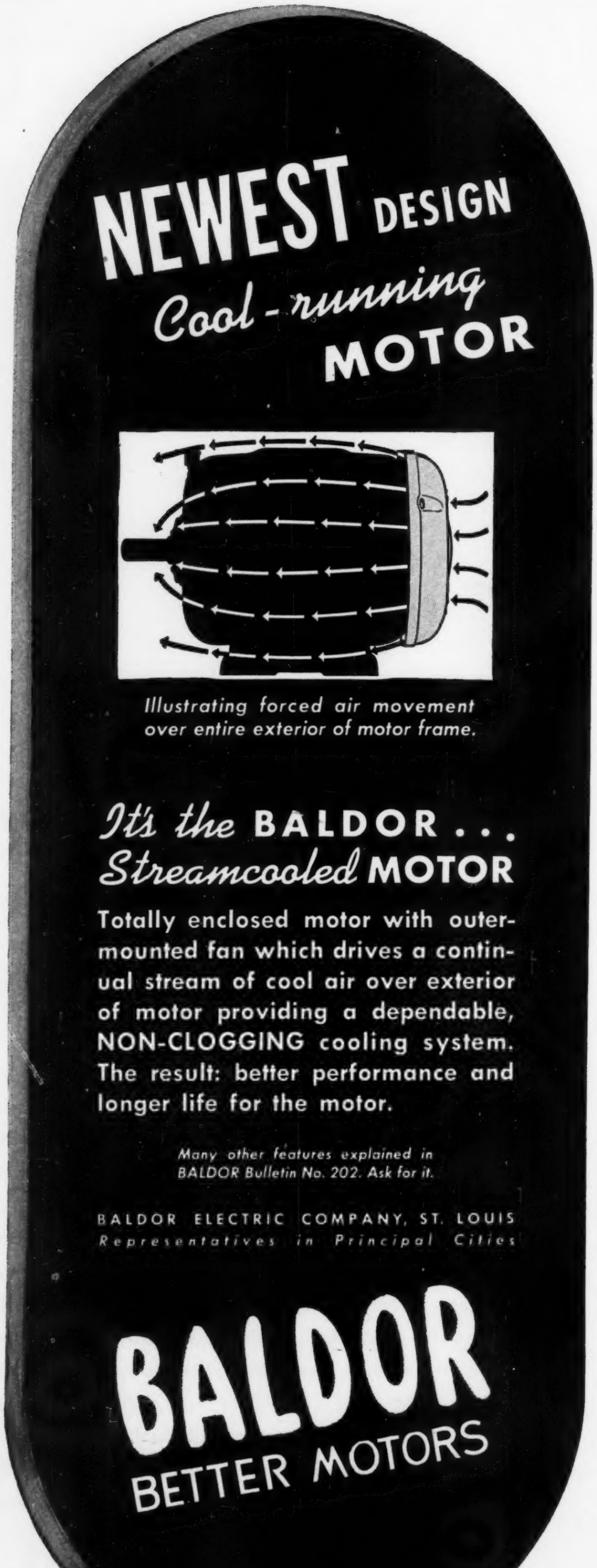


CLAMPING with HYDRO-POWER

A prominent manufacturer★ of die castings asked HYDRO-POWER to design and build an efficient clamping unit for his machine. Above is the result . . . a straight line hydraulic clamp with a self-contained operating system, insuring positive clamp and automatic adjustment for die thickness . . . PLUS . . . more production per hour, fewer rejects, automatic operation, larger castings with same die space, fully adjustable clamping pressure and stroke, with automatic slow-down prior to die contact. ● This is one of many applications where HYDRO-POWER has accomplished more efficient production. Call in a qualified HYDRO-POWER engineer to assist in designing hydraulics into your machine. ★Name upon request.

HYDRO-POWER SYSTEMS, INC.
Mount Gilead, Ohio, U. S. A.

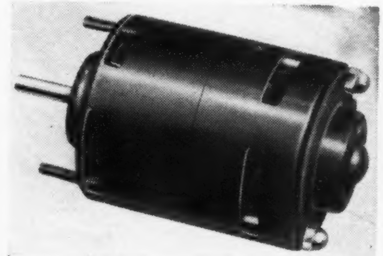
"First Cost is Investment • Maintenance is Expense"



tions per minute. Motor armature and pump impeller are on a single shaft.

Offer Small Universal Motor

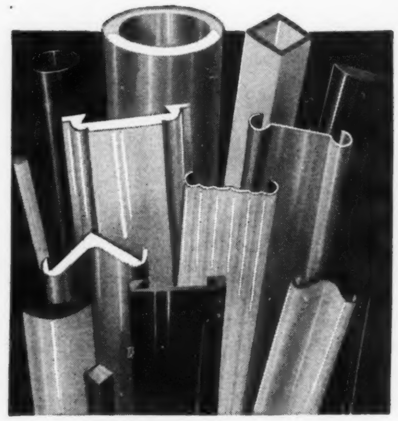
SUITABLE for operation in horizontal or vertical positions and for use in portable or stationary equipment, the 115-volt universal motor announced



by Delco Appliance Division of General Motors Sales Corp., Rochester, N. Y., develops from 1/100 to 1/200 horsepower depending on speed and type of service. It measures 2⅜ inches by 4 inches and has a ¼-inch shaft, operating at 3000 to 8000 revolutions per minute. The motor may be wound for either 6 or 12-volt operation. In its present form it lends itself readily to use on equipment such as jewelers' tools, small picture projectors, cooling fans and various miniature types of apparatus.

Plastics Extruded Continuously

EXTRUDED plastics in continuous lengths and many cross-sectional patterns are announced by the Chicago Extruded Plastics Co., Chicago. Indus-



trial plastics are emphasized which are impervious to gasoline,oils and other organic substances. Among the advantages claimed for the new method of extrusion developed by the company is low die cost, making practical comparatively short runs.

Time Delay Switch Is Announced

TIME delay switch announced by Betts & Betts Corp., New York, is designed for laboratory and industrial application in conjunction with magnetic

(*Continued on Page* 90)

FIBRO FORGED SCREWS

TRADE MARK

Completely Cold Forged

by

HOLO-KROME

Presenting a "Three Fold"

Opportunity to Designers

Save **WEIGHT—SPACE—TIME**

Incorporate Holo-Krome FIBRO FORGED Socket Screws in the Design of new labor-saving equipment, special purpose machines and in the re-designing of standard machines to increase the efficiency of their production. You will save *Weight, Space and Assembly Time* by using these Precision Made Completely Cold Forged Screws, a fact, daily proven by the experiences of Designers in thousands of plants throughout the country.

Write "HOLO-KROME" into *your* specifications—

Guaranteed

UNFAILING PERFORMANCE

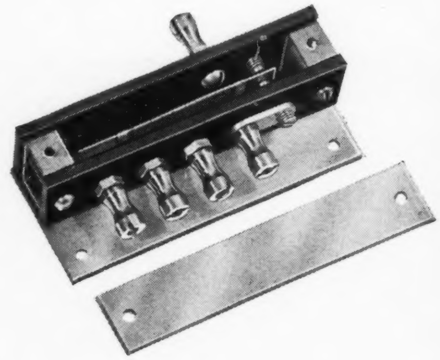
HK HOLO-KROME

fibro forged

THE HOLO-KROME SCREW CORP. **SOCKET SCREWS** HARTFORD, CONN. U.S.A.

(*Continued from Page* 86)

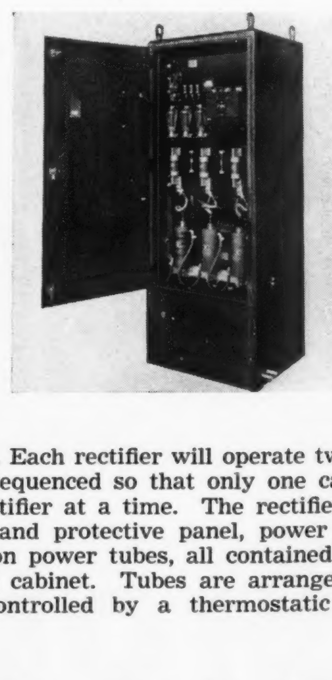
relays. It is generally used where temperature changes are fairly constant and a slight variation in timing is not detrimental. Special compensating



switches are provided where accurate delay is essential. Provided with four terminals, two for the heater coil and two for the main curcuit, the switches are adjustable within the limits of one second to five minutes. They are available in immediately or not-immediately recycling types, normally open or normally closed.

Rectifier Supplies Spot Welders

FOR supplying required direct current to magnetic energy-storage type spot welders, an ignition spot welding rectifier is announced by the Westinghouse Electric & Mfg. Co., East Pittsburgh. Rectifiers are available in two capacities, one for use with 40-kilowatt spot welders, the other for 120-kilo-



watt units. Each rectifier will operate two welders if they are sequenced so that only one can be loaded on the rectifier at a time. The rectifier consists of a control and protective panel, power transformer and ignition power tubes, all contained in a forced-ventilation cabinet. Tubes are arranged for water-cooling, controlled by a thermostatic water flow switch.

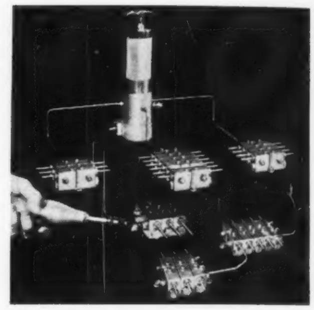
Molding Material Developed

TO BE known as Durez 775 Black, a new general-purpose phenolic molding compound is announced by Durez Plastics & Chemicals Inc., Walck road, North Tonawanda, N. Y. This compound was

developed to make available a material with a wider range of application, a lower water absorption, and a slightly higher flexural and tensile strength, with a heat resistance of 400 degrees Fahr. Its molding characteristics are excellent, delivering a smoother, more lustrous finish than the average general-purpose material.

Lubricates by Nonreversing System

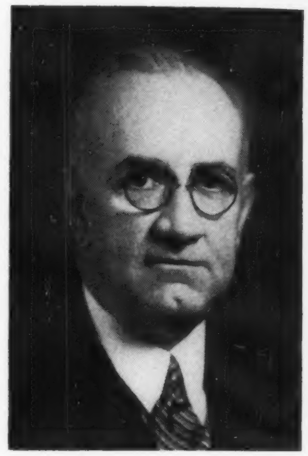
SINGLE-PIPE, nonreversing system of lubrication introduced by Trabon Engineering Corp., 1814 East Fortieth street, Cleveland, makes positive forced feed centralized lubrication almost universally applicable. The new system is composed of the company's non-reversing single-inlet multi-outlet distributor feeder and the new "MP" series variable feed multi-outlet pump. This improved feeder or distributor consists of a bank of three or more sections, each discharging a known and measured quantity of lubricant alternately through one or two discharge outlets which are direct connected to bearings. The capacity or volume of the sections may vary from .005 cubic inch to .035 cubic inch. By selecting the proper number and capacity of sections, and supplying the proper amount of lubricant to the inlet, a single distributor discharging progressively through one outlet after another, will deliver the desired amount of lubricant to all connected bearings. Either oil or grease may be used as a lubricant. The distributor must discharge lubricant from every outlet or the operator receives immediate warning. The new "MP" multi-outlet pumps, designed specially for use with these nonreversing systems, are available in three sizes having different reservoir capacities. They are available for use with any chain, gear, V-belt, or direct motor connection, or can be furnished with overrunning clutch if the drive is taken from a cam or some oscillating part.



Cord Belt Meets Conveyor Needs

COMBINING flexibility of cord construction with ability to hold metal fasteners to a degree previously approached only by fabric construction, a new conveyor belt has been introduced by The Manhattan Mfg. Division, Passaic, N. J. Construction of the conveyor belt is of a different type of cord, neither standard weave nor standard cord design. It was developed particularly to meet requirements of conveyor belt use, and in its present form is not intended for other uses such as for transmission belting.

MEN *Of Machines*



ONE of the pioneers in the aviation industry, Col. Carl M. Tichenor has recently been appointed executive vice president and general manager of Doyle Machine & Tool Corp., Syracuse, N. Y. With his experience dating back to early developments in the aircraft industry (in 1907) when as an engineer he designed and built aviation motors for the Chinese government, he is well qualified to take over the duties of his new position with a company engaged in aeronautical engineering work, and in production of precision tools and machinery.

In 1912 Colonel Tichenor was named chief engineer for Gray & Davis Inc. For the following fifteen years he was identified with the Everitt, Metzgar, Flanders interests as division chief engineer of Everitt and Maxwell Motor Companies, Rickenbacker Motor Co., and Universal Pressed Steel Co. During 1928 and 1929 he was vice president of Freshman-Freed Eisman Radio Corp. and then became general manager of Muskegon Works (musical division) of Brunswick Balke Collander. He continued in this capacity until he became active in industrial engineering work in the Michigan area.

CHIEF engineer of Enterprise Engine & Foundry Co., San Francisco, since 1932, Hans Bohuslav has been named vice president in charge of engineering. Since joining Enterprise Mr. Bohuslav has been closely connected with a number of new and interesting designs such as multiple cylinder stationary and

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Variable speed control units that are creating records of *efficiency that will never be forgotten,* for the machines on which they are installed.

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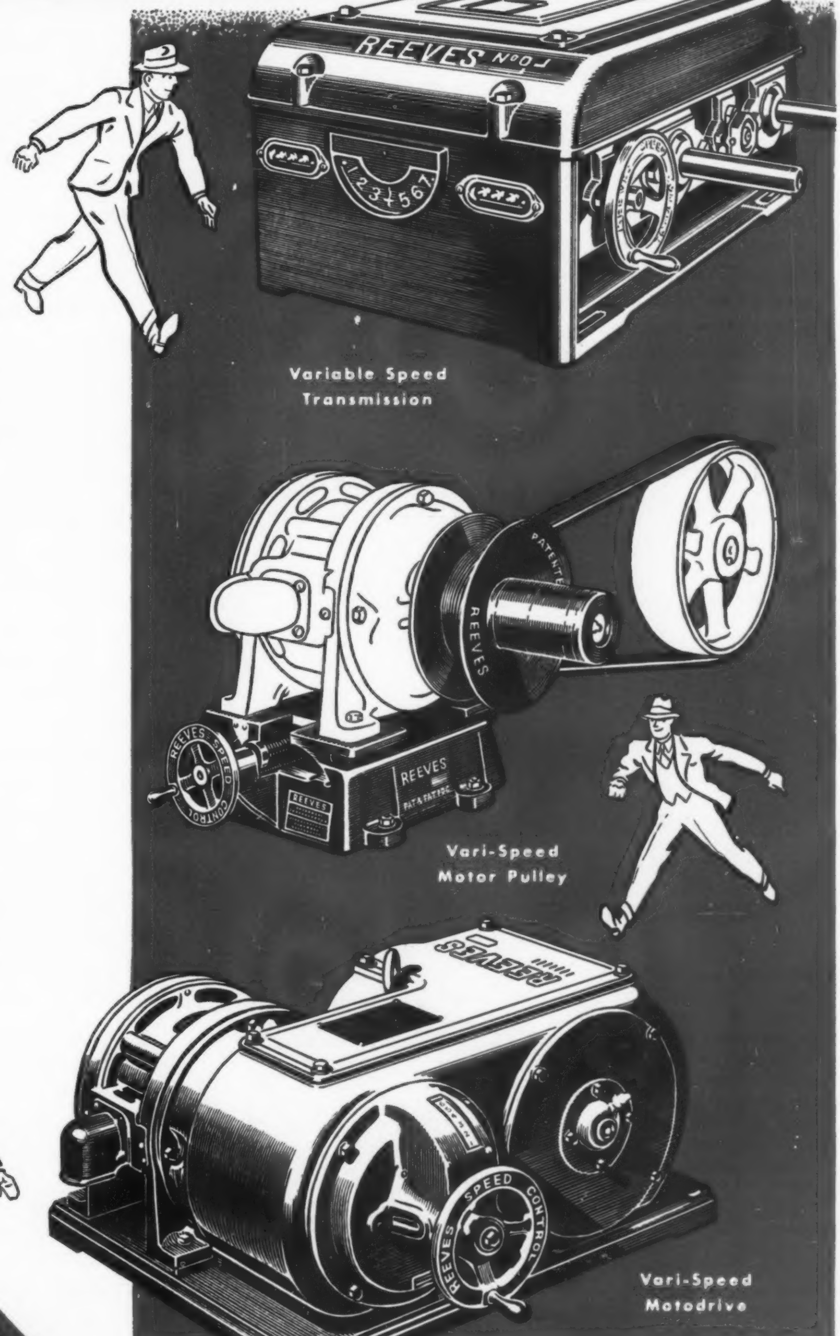

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marine diesel engines. These engines were of integral block design and utilized standardized construction while adapting materials and special features to fit the requirements of unusual service conditions. Lately he has been associated with the development of special design of high output, low weight units to meet demands of national defense.

His training consists of a combination of theoretical and practical work, university courses and machine shop practice in United States and abroad. Prior to becoming employed by Enterprise, his experience included that of design and development of experimental diesel engines, fuel injection equipment and special apparatus. He has also had several years in merchant marine service.

♦

DONALD C. BAKEWELL, vice president of Union Steel Castings division of Blaw-Knox Co., Pittsburgh, has been re-elected president of the Steel Founders' Society.

♦

PAUL W. ARNOLD, a mechanical engineering graduate and since engaged in application engineering work for Reliance Electric & Engineering Co., has been named manager of machinery design applications. ROSCOE H. SMITH who from 1932 has devoted all of his time to application engineering, now becomes manager of applied engineering.

♦

E. S. CHAPMAN, vice president and assistant general manager of Chrysler's Plymouth plant, has been loaned to the National Defense commission where he will devote his efforts to assisting in organizing and directing the vast industrial machinery of the country.

♦

HALL L. HIBBARD, vice president and chief engineer of the Lockheed Aircraft Corp., has been elected vice president of the Institute of Aeronautical Sciences. FRANK CALDWELL of United Air Lines was elected president.

♦

DR. EDWARD E. MINOR JR., research and development engineer of Glenn L. Martin Co., Baltimore, has been appointed chairman of the new air transportation committee of the American Institute of Electrical Engineers.

♦

PAUL GILLAN, formerly assistant chief engineer and automotive engineer, Lycoming Mfg. Co., Williamsport, Pa., has joined Aluminum Industries Inc. as a member of its automotive and industrial consultation staff at Cincinnati. He will work with the engineering staffs of automotive, aircraft and industrial companies on parts, casting problems.

♦

GUY W. VAUGHAN, who won distinction as a designer and engineer in the automotive industry, and who is president of the Curtiss-Wright Corp.

(*Continued on Page* 100)

(Concluded from Page 96)

and its subsidiary, Wright Aeronautical Corp., has been elected a director of Western Electric Co. Mr. Vaughan entered aviation in 1917 as quality engineer of Wright-Martin Co. During his vice-presidency of the Wright Aeronautical Corp. he supervised development of the Wright Whirlwind engine, first of the now popular air-cooled radial engines. He was elected president of Wright Aeronautical Corp. in 1929, and of Curtiss-Wright in 1935.

♦

L. O. MYHRE, previously manager of engineering and manufacturing for Westinghouse X-ray Co., has been appointed manager of manufacturing in the radio division of Westinghouse Electric & Mfg. Co., Baltimore.

♦

EDGAR F. WENDT, president, Buffalo Forge Co., Buffalo, has been elected president of the National Association of Fan Manufacturers.

♦

GEORGE H. TWENEY, recently in charge of general engineering for the Atlantic division of Pan-American Airways, has become an instructor in aeronautical engineering at the University of Detroit. A graduate of this university in 1938, Mr. Tweney will assist GEORGE J. HIGGINS, who became director of the aeronautical engineering department recently when Peter Altman left the university.

♦

D. W. LEE, graduate of Massachusetts Institute of Technology and former member of the Chrysler engineering department, will co-ordinate development for the entire line of Dodge job-rated trucks, both gas and diesel powered.

♦

FRANK D. NEWBURY, formerly manager of the emergency products division of Westinghouse Electric & Mfg. Co., has been named vice president. Since he enrolled in the company's training course in 1903, he has served in the following capacities: Design engineer, and after various promotions, assistant to vice president in 1935, and director of the new products division in 1938. A. C. STREAMER, formerly general manager of East Pittsburgh works of the company, has also been appointed vice president. Still another new vice president, R. A. MCCARTY, who has been with the company since 1903, was successively design, section and division engineer, manager of the generator engineering department, manager of the small motor plant at Lima, O., and manager of the steam division.

♦

B. F. BOWER has resigned as chief engineer of Howell Co., St. Charles, Ill., to organize a company which will shortly begin production of tube fabricating machines. He has spent many years in designing and perfecting tube fabricating equipment and other metalworking machines for making tubular steel.

Challenges to Design Result in Progress

(Concluded from Page 45)

dials are journaled in small bronze bearings. Here, as in the case of the keyboard, the dials are positioned in the half bearings of one of the U-sections where they are easily maintained until the other is bolted into place.

The two U-sections of the carriage are extruded duralumin for lightness and saving in machining time since this extruded stock comes within the tolerances necessary for this type of construction. The end plate for the carriage as well as the base of the machine are die-cast zinc which, with their numerous bosses and screw holes, are cast within the necessary tolerances thereby saving hours of machining, drilling and countersinking time.

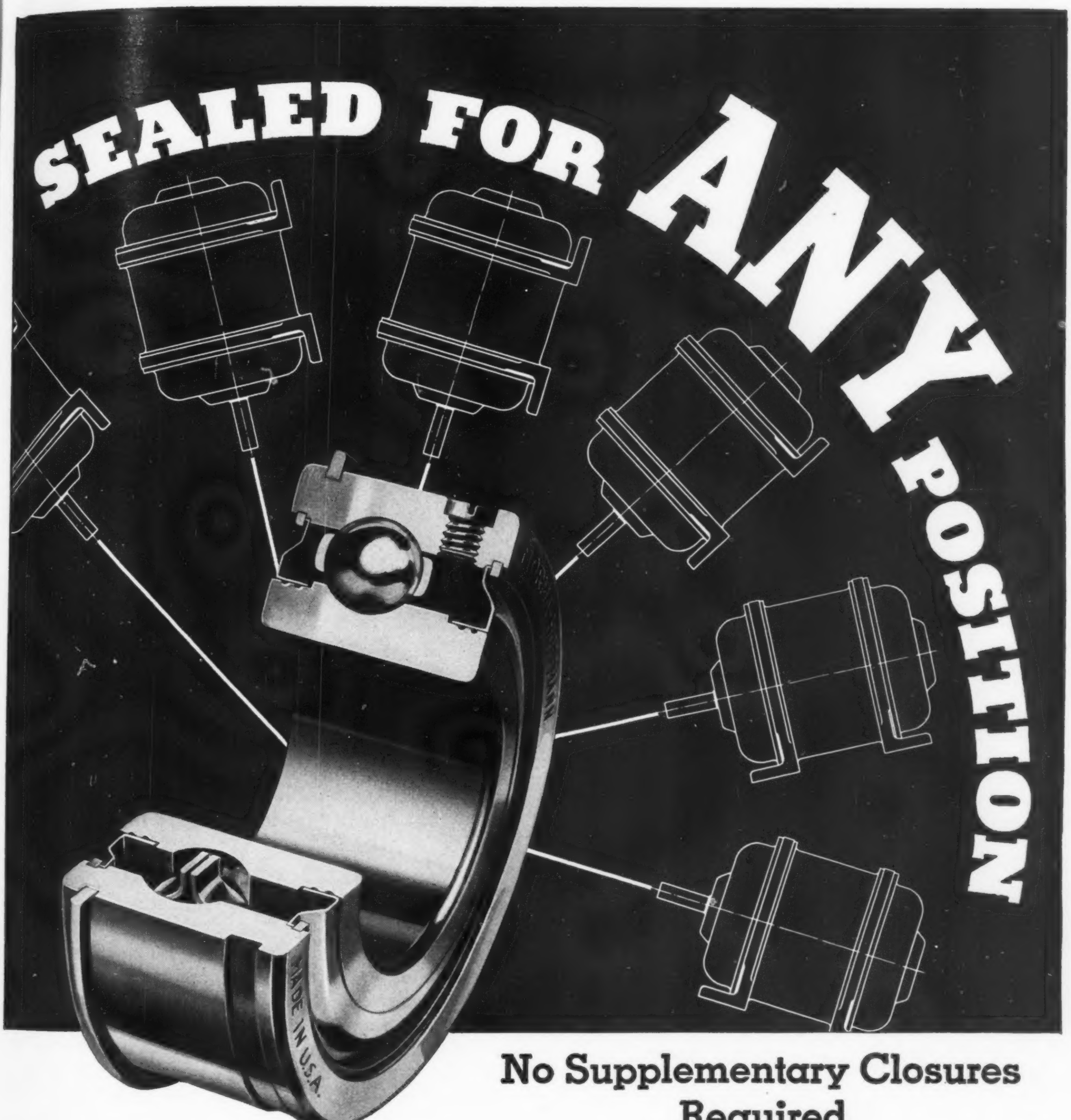
Lubricant Adheres to Parts

In all types of high speed office machines where working mechanisms, bearings, etc., may become warm through continuous running, and grease and oil cups are too cumbersome to be adopted, it is difficult to maintain proper lubrication. Those who have experienced this difficulty may be interested to know that this problem has been solved by a product called Lubriplate. In addition to high viscosity, this product has adhesive qualities which makes it adhere to moving parts, gears and bearings in spite of heat, centrifugal force or atmospheric conditions.

Although external appearance of office machines has been considered secondary to trouble-free operation and the longevity of the operating mechanism within, any machine should be outstanding in appearance in its own right. With this in mind, the covers of the calculator were redesigned to eliminate all sharp corners and give the effect of a streamlined unit regardless of the angle from which it is viewed.

Since psychologists have found that harsh colors also have an unfavorable effect on the eyes and nerves of an operator, the most pleasing combination of colors possible has been adopted. This combination comprises three tones of soft green for all operating and setting buttons against a background of gray. The only departures are small chrome-plated name plates and corner pieces which are not seen from the operating position. One tiny red arrow adds a dash of color to break the field of gray and green and tends to focus the operator's attention to the units order of the keyboard which is the point from which all decimal places are gaged.

With every new development speed and simplicity are improved. The constant challenge that faces the designers seems to result unfailingly in further progress. This happy condition should always prevail and as a result, better, finer, and more automatic machines will continue to be developed.



No Supplementary Closures Required

Regardless of the angle of the shaft—horizontal, vertical, or in between—the "CARTRIDGE" BALL BEARING seal, with its long flange, minute clearance, recessed inner ring construction and multiple grease grooves, KEEPS THE GREASE IN AND THE DIRT OUT.

Thoroughly tested in our own laboratory over a protracted period, and in the field under continuous duty, the "CARTRIDGE" BALL BEARING has demonstrated not alone its value but its SUPERIORITY.

One of the largest motor manufacturers equipped a vertical motor with "CARTRIDGE" BALL BEARINGS and after running it on test over four years, nine hours a day, five days a week, writes:—"*we have added no grease to either bearing and have observed no leakage*". This test motor is still running, with the same results.

Adopt the fully-sealed "CARTRIDGE" BALL BEARING as absolute protection against leakage, neglected lubrication, and dirt and grease contamination. It also eliminates numerous supplementary closure parts, machining operations and variables, and speeds up production. Moreover, it has convenient regreasing and inspection features.

Write for the Catalog. Let our engineers work with you.

NORMA-HOFFMANN BEARINGS CORPORATION, STAMFORD, CONN., U. S. A.

Designing Snap Rings

(Continued from Page 51)

the ring is compressed so that the hooks touch, the ring will not take an appreciable permanent set.

It is important that the distance across corners of square or rectangular wire be kept at the maximum, that is, the radius of the corners should be as small as possible. For example, on wire .062-inch square the corner radii should be in the order of .005-inch with the distance across corners approximately .083 plus or minus .003. The snap ring manufacturer should be consulted as to the minimum corner radius that is practicable.

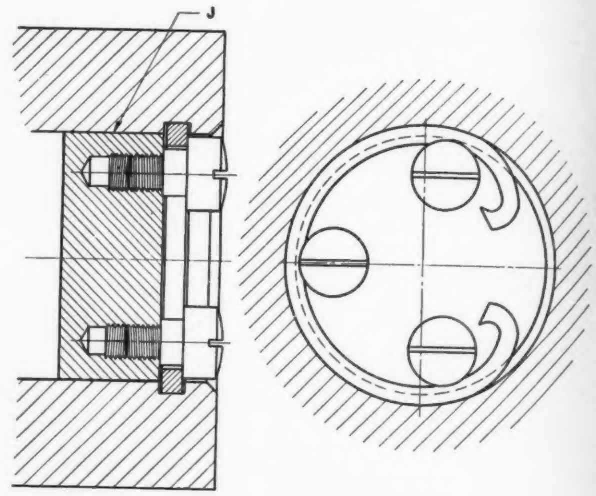


Fig. 8. Fillister head screws tapped into plug provide a third alternative for snap ring locking. Special screw shank is flush with plug; head does not clamp the ring

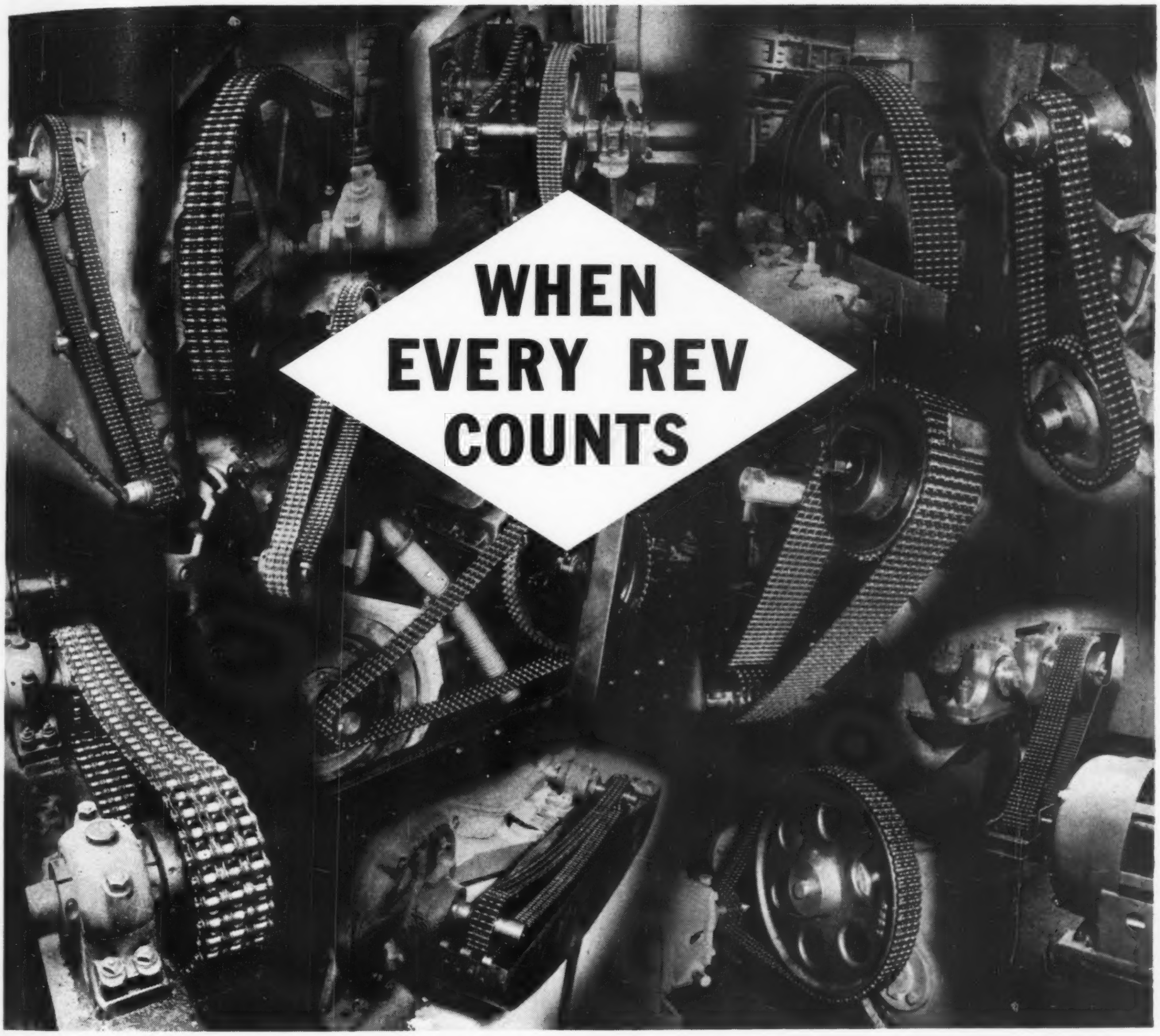
External rings made of square stock should have a groove depth of one-fourth the radial dimension of the ring cross section; those made of round stock, a groove depth of one-third the diameter.

Internal rings made of square or rectangular stock have a groove depth of approximately one-half to two-thirds the ring cross section. Those made of round stock have a groove depth of about one-third the ring cross section. Excessive chamfers on the edges of internal grooves should be avoided.

For rings made of round wire the radius of the groove equals the radius of the wire cross section, and for rings of square or rectangular stock, the width S and groove width W are as follows:

If S = .02 to .03, W = S + .004
.03 to .04, W = S + .005
.04 to .05, W = S + .006
.05 to .10, W = S + .007
.10 to .20, W = S + .008

TOLERANCES: Maximum tolerance for inside diameter on external rings and outside diameter on internal rings should be plus or minus .01 per inch of diameter. The gap dimension should be held to +1/64,—0 per inch of diameter. Commercial tolerances on wire width and thickness are plus or minus 2 per cent. Groove width may be held to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plus or minus .005. Tolerance for groove diameter should be +0, —.005 per inch of diameter.

Plating of Rings: There should be no plating on internal snap rings. On external rings highly stressed or of small cross section such as 1/16-inch square, cadmium plating should be avoided because the embrittlement, despite subsequent baking operations, tends to precipitate breakage.

In addition to the normal design considerations there are conditions where, for reasons of safety, it is especially desirable to protect against damage due to the remote possibility of ring breakage.

In *Fig.* 5 the plug J has diameter K flatted to clear the ring hooks. When assembling the ring the plug is pushed inward against the mechanism spring so that the diameter K clears the inner edge of the groove and, after the ring has been seated in the grove, the plug J is released. The diameter K then securely locks the ring in place. The projection of diameter K can be equal to the width of the ring but one half of the ring width suffices.

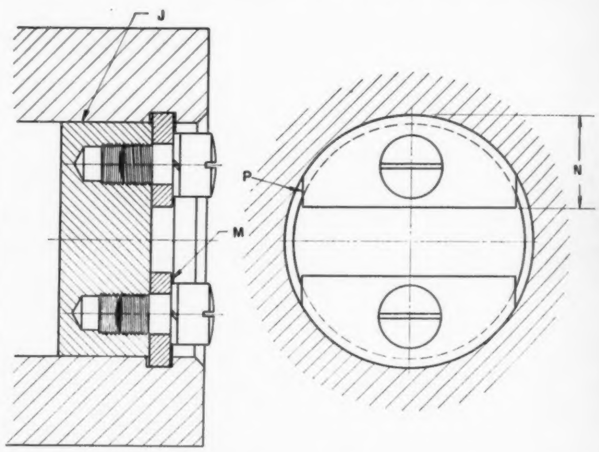

Fig. 9. Where thrust loads are extremely high, plate locks may provide a satisfactory solution

For conditions where plug J is shouldered as shown in *Fig.* 6, the dee washer L can be used. *Fig.* 8 illustrates the application of shoulder screws as the locking means where it is necessary to locate the plug J axially. Since lock washers are impractical for this shoulder screw application, the screw heads may be drilled and safety wired.

Where the thrust loads on the plug J are of such magnitude as to render a snap ring impractical, the plate locks M in *Fig.* 9 can be used to good advantage. Since the locks M must be moved toward the center for removal, the width N and the amount of flat at P depends upon the groove dimensions and other characteristics.

For locking external snap rings, as in *Fig.* 7, the inner end of part S is counterbored to retain the ring and the other end utilizes the washer T which is staked at intervals as at U after the ring is assembled. Application shown on the right of *Fig.* 7 is similar except a solid ring, V, is pressed over the end ring.

The chart, *Fig.* 4, is supplied through courtesy of Barnes-Gibson-Raymond, Division of the Associated Spring Corp.

Theory of Elasticity in Practical Design

(*Continued from Page* 59)

this and $E_b = 29 \times 10^6$, $\lambda_b = .3$ in Equation 81

$$q = \frac{.001 \times 14 \times 29 \times 10^{12}}{(7.53 - .35)\ 29 \times 10^6 + (1.667 + .3)\ 14 \times 10^6}$$

$$= 1725 \text{ lbs. per sq. in.}$$

From Equations 74 and 75, the stress in the bushing is 12960 pounds per square inch at r and 14700 at r_a, both compression. The stress in the steel hub is 2870 pounds per square inch tension at r. Equations for u_a and u_b show that the bushing closes in at the common radius 88.3 per cent of the interference and the steel hub opens up only 11.7 per cent. From Equation 80 the inside of the bushing closes in 91.9 per cent of the interference[1]. Thus it is seen that the common assumption that the bushing closes in practically the entire amount of the interference will, in the usual case, be not far from the truth. Notice also that the hole in the bushing has closed in more than the outside radius. This condition will exist from $k = 1$ down to $k = (1 - \lambda)/(1 + \lambda)$. Below this the inside closes in less, moving not at all at $k = 0$. This condition does not exist for the external member, the outside always opening less than the inside.

If the material is the same in both members, Equation 81 reduces to

$$q = \frac{\epsilon E}{K_a + K_b} \qquad (82)$$

The changing of the bronze bushing in the preceding example to steel will increase q to 3,150 pounds per square inch and the maximum stress in the bushing to 26,850 pounds per square inch.

If a compound cylinder is subjected to an internal pressure q_a at r_a, the displacement u_a at r for the inside cylinder will be given by

$$u_a = -\frac{r}{E_a}[\ (K_a - \lambda_a)q - (K_a - 1)q_a]$$

and as before $u = u_b - u_a$. If the material is the same, the increase Δq in the contact pressure q, is given by

$$\Delta_q = \frac{K_a - 1}{K_a + K_b} q_a \qquad (83)$$

The stresses from $(q + \Delta q)$ may now be found from Equations 72 to 76.

For the force fit of a ring on a solid member, returning to Equation 79 and substituting the stress

(*Continued on Page* 116)

[1] See also some specific examples of another type discussed by Clinedinst in MACHINE DESIGN, June, 1936, Pages 43 to 45.


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SURE FLOW
"LOGAN"
LOGAN
HYDRAULIC

(Continued from Page 110)

value for a solid disk. Equation 71 gives for the de-formation of the disk

$$u_\rho = -\frac{(1-\lambda)\,q\rho}{E} \quad \ldots\ldots (84)$$

and, using the notation of before with a applying of the disk,

$$u_a = -\frac{(1-\lambda_a)\,q\,r}{E_a} \quad \ldots\ldots (85)$$

As before, $u = u_b - u_a$. Substituting and solving for q

$$q = \frac{\epsilon\, E_a\, E_b}{(1-\lambda_a)\,E_b + (K_b + \lambda_b)\,E_a} \quad \ldots\ldots (86)$$

In a shrink fit between a shaft and the hub of a gear or wheel, for instance, the inner member is many times longer than the outer. However, since the stress sum is constant over the cross section the extension beyond the fit is moved along the Z axis without disturbing the stress distribution under the fit. Therefore the shaft may be treated as a disk of the same thickness as the length of the hub.

However, in such case there is an important "stress concentration" in the Z direction at the end of the fit. The surface of the shaft is compressed under the hub, whereas immediately beyond it has its original position. This results in an increase in the contact pressure at the ends and a surface stress in the Z direction. If a bending load is applied to the shaft at the same time the stress increase may be considerable. The reader is referred for further discussion of this subject to an article[2] on fits in railroad axles by Horger and Buckwalter of The Timken Roller Bearing Co. The reduction of this fit end condition by a groove in the shaft is there described. Also, O. J. Horger in another paper[3] describes the relief of this condition by a groove in the face of the hub and a raised portion on the shaft. Photoelastic pictures are given in both cases, the latter paper giving a photo of the setup.

As an example of a ring and disk, consider a cast iron hub with an effective external radius of 4 inches shrunk over a 4-inch diameter steel shaft with an interference of .001-inch per inch. Taking $E_b = 12,$-$000,000$ and $\lambda_b = .25$

$$q = \frac{.001 \times 29 \times 12 \times 10^{12}}{(1-.3)\,12 \times 10^6 + (1.667 + .25)\,29 \times 10^6}$$

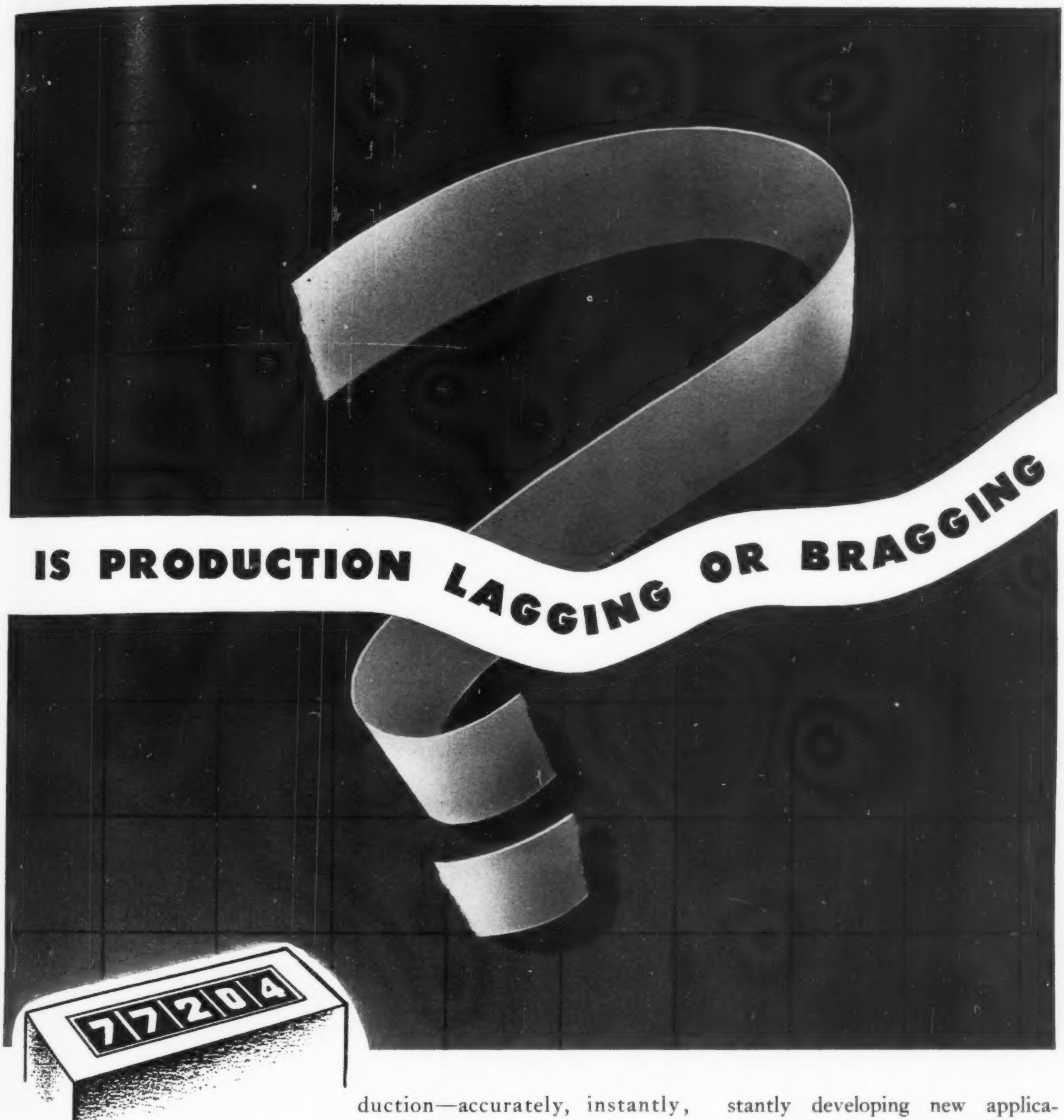
$$= 5450 \text{ lbs. per sq. in.}$$

and the stress in the hub is, from Equation 74, 9,050 pounds per square inch tension.

The stress in the hub is given directly as

$$S_\theta = K_b q = \frac{\epsilon\, E_a\, E_b\, K_b}{(1-\lambda_a)\,E_b + (K_b + \lambda_b)\,E_a} \quad \ldots (87)$$

[2] *Metal Progress*, February, 1941.
[3] *Journal of Applied Physics*, July, 1938.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and the "equivalent shear failure" stress, Q, from Equation 76

$$Q = (K_b + 1)q = \frac{\epsilon E_a E_b (K_b + 1)}{(1 - \lambda_a) E_b + (K_b + \lambda_b) E_a} \quad \ldots\ldots (88)$$

Both of these equations have a limit at the point where the hub becomes a thin hoop, for which $k_b = 1$ and K_b becomes infinite. Substituting in Equations 87 and 88

$$S_\theta = Q = \epsilon E_b \quad \ldots\ldots (89)$$

The other limit of Equations 87 and 88 is when the hub becomes infinitely large, for which $k_b = 0$ and $K_b = 1$. This is in all cases a lower limit for 87. However, 88 will be a lower or upper limit depending upon the relative values of the elastic constants. When $E_b/E_a = (1 - \lambda_b)/(1 - \lambda_a)$ Q at the inside of the hub will remain independent of the hub thickness. This will, of course, be the case if both are of the same material. For the ratio of the elastic moduli more than this, Equation 89 will be the upper limit; otherwise (and this is more apt to be the case) it will be the lower.

For hub and shaft of the same material Equations 87 and 88 become

$$S_\theta = \frac{1}{2} \epsilon E (1 + k_b^2) \quad \ldots\ldots (90)$$

$$Q = \epsilon E \quad \ldots\ldots (91)$$

Values of S_θ and Q are given in the Table for three particular cases of a cast iron or steel hub on a steel shaft, using the values of the elastic constants previously given.

Stress in Bore of Hub on Steel Shaft

Multiply values by 1000ϵ

Type of Hub		Thin Hub	Average Hub	Thick Hub
	k =	1	½	0
Cast Iron Hub	S_θ	12,000	9,050	7,800
	Q	12,000	14,500	15,600
	ASA	.....	10,432	.....
Steel Hub	S_θ	29,000	18,100	14,500
	Q	29,000	29,000	29,000
	ASA	.....	29,000	.....

The data noted "A.S.A." is taken from the American Standards Association standard B4a-1925 on Fits, and is stated to be based on $k = ½$. Values of E and λ used are not reported.

This discussion of force fits, compound cylinders, etc., could be continued to many other interesting problems, such as the effect of relieving the stress by a hole in the shaft, the force required to assemble fits, etc. However, the discussion has been sufficient to enable the reader to develop the equations to fit his particular problem, and to direct his

attention to the wide applicability of the solution of thick cylinders. The reader is further referred to a discussion[4] by Horger and Nelson on press fits, and to the bibliography appended thereto.

Succeeding parts of this series will continue the development of the ring Equations 68 and 69 to the remaining case where D is not equal to zero—that of the curved beam under uniform bending. Also, other cases of curved beams will be discussed.

[4] *Journal of Applied Mechanics,* December, 1937 and March, 1938.

Meetings and Expositions

May 9-11—

Laundry and Cleaners Allied Trades association. Educational clinic and exhibition to be held at Grand Central Palace, New York. Rodger R. Jackson, 95 Liberty street, New York, is manager.

May 11-15—

National Electrical Manufacturers association. Spring conference to be held at Homestead hotel, Hot Springs. R. J. Blais, 155 East Forty-fourth street, New York, is convention manager.

May 12-13—

Society of Automotive Engineers. National production meeting to be held at Schroeder hotel, Milwaukee. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary.

May 12-15—

American Foundrymen's association. Forty-fifth annual convention to be held at Pennsylvania hotel, New York. N. F. Hindle, 222 West Adams street, Chicago, is assistant secretary.

May 19—

American Welding Society. West coast district meeting to be held in Los Angeles, in conjunction with the Western Metals Congress and Exposition. Additional information may be obtained from M. M. Kelly, 33 West Thirty-ninth street, New York, secretary.

May 19-21—

American Institute of Chemical Engineers. Semiannual meeting to be held at Edgewater Beach hotel, Chicago. Additional information may be obtained from S. L. Tyler, 50 East Forty-first street, New York, secretary.

May 26-27—

Stoker Manufacturers association. Annual meeting to be held at the Greenbrier, White Sulphur Springs, W. Va. Marc G. Bluth, 307 North Michigan avenue, Chicago, is secretary.

June 1-6—

Society of Automotive Engineers. Summer meeting to be held at the Greenbrier, White Sulphur Springs, W. Va. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary.

June 16-20—

American Institute of Electrical Engineers. Annual summer convention to be held at Toronto. H. H. Henline, 33 West Thirty-ninth street, New York, is national secretary.

Business and Sales Briefs

SINCE 1938 sales manager of Dumore Co., Racine, Wis., R. L. Hamilton has been elected vice president of the company. He has been with the company since graduating in 1934, and was advertising manager from 1935 to 1937.

♦

Robert F. Moody of the sales engineering staff of Wolverine Tube Co., Detroit, has been moved to the home office at Detroit. Mr. Moody has traveled in the middlewestern states for the past twelve months.

♦

Due to increased activities the Indianapolis sales office of Cutler-Hammer Inc. has been moved to 241 North Pennsylvania street, Room 316. G. E. Hunt, who has had over 20 years of experience in the application of controls, continues in charge of the Indianapolis office.

♦

Percy Jenkins has resumed all duties which he previously held as district sales manager for the New England territory for Wickwire Spencer Steel Co., New York. He is located at 80 Webster street, Worcester, Mass.

♦

Met-Alloys Inc., Chicago, has recently been organized with offices at 332 South Michigan avenue, and has equipped its plant at 425 West Chicago avenue, East Chicago, Ind., to produce nickel and copper alloys in shot form, as well as phosphor copper and other metals. George Birkenstein is president, and George Birkenstein Corp., Chicago, will handle all purchases and sales for the new company.

♦

Plant facilities of the Trumbull Electric Mfg. Co., Plainville, Conn., have been increased by the purchase of two new factories. Equipment from its Ludlow, Ky., plant has been transferred to a larger factory in Norwood, O. The company has also purchased an additional factory in Plainville, Conn., to be known as the West Plant.

♦

Organization of the American Nickel Alloy Mfg. Corp. has been announced by the parent company, Anglo-American Metal & Ferro Alloy Corp., which has moved from 200 Broadway to 50 Church street, New York. The new company, which will manufacture nickel alloys and allied products, will also be located at the new address. Alfred J. Brunebaum is president of both firms.

♦

Chromium Corp. of America, Chicago, has completed construction of a one-story addition to its plant. According to a recent announcement, the company is operating 24 hours a day, seven days a week, and is engaged in considerable defense work.

♦

Plans for reopening and renovating its former motor plant in Taunton, Mass., have been completed by General Electric Co., in order to expand its facilities for manufacturing molded plastic parts. This plant, which increases the number of G-E plastics plants to five, will be in operation by July.